BASIC HYDRAULICS

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PREFACE

This book was written primarily for the student Gunner's Mate, and was designed for use in Naval Gunner's Mate Schools which provide a course in basic hydraulics.

Normally the basic hydraulics course is covered in a relatively short period, and is followed by other courses leading to a more specialized study of ordnance hydraulic systems. It is highly important that the Gunner's Mate have a sound knowledge and understanding of basic hydraulic principles before he starts his specialized studies. Because the students who take the basic course vary in their experience with hydraulic systems, there is every reason for simplicity and clarity in this presentation.

It was realized that knowing how a system works is just as important as practical acquaintance with the system. Neither form of experience is a substitute for the other. An effort has therefore been made not only to describe but to explain.

It is nevertheless a fact that much remains unexplained in this book, and must be accepted as part of the body of scientific thinking. There are no mathematical demonstrations, and very few formulas. Wherever possible the principles of hydraulics have been shown to be consistent with everyday knowledge and experience.

It is believed that this volume will be of value to all branches of the service which employ hydraulic equipment. Acknowledgement is made of the cooperation of the Bureau of Ships and the Bureau of Ordnance in the preparation of this manual.

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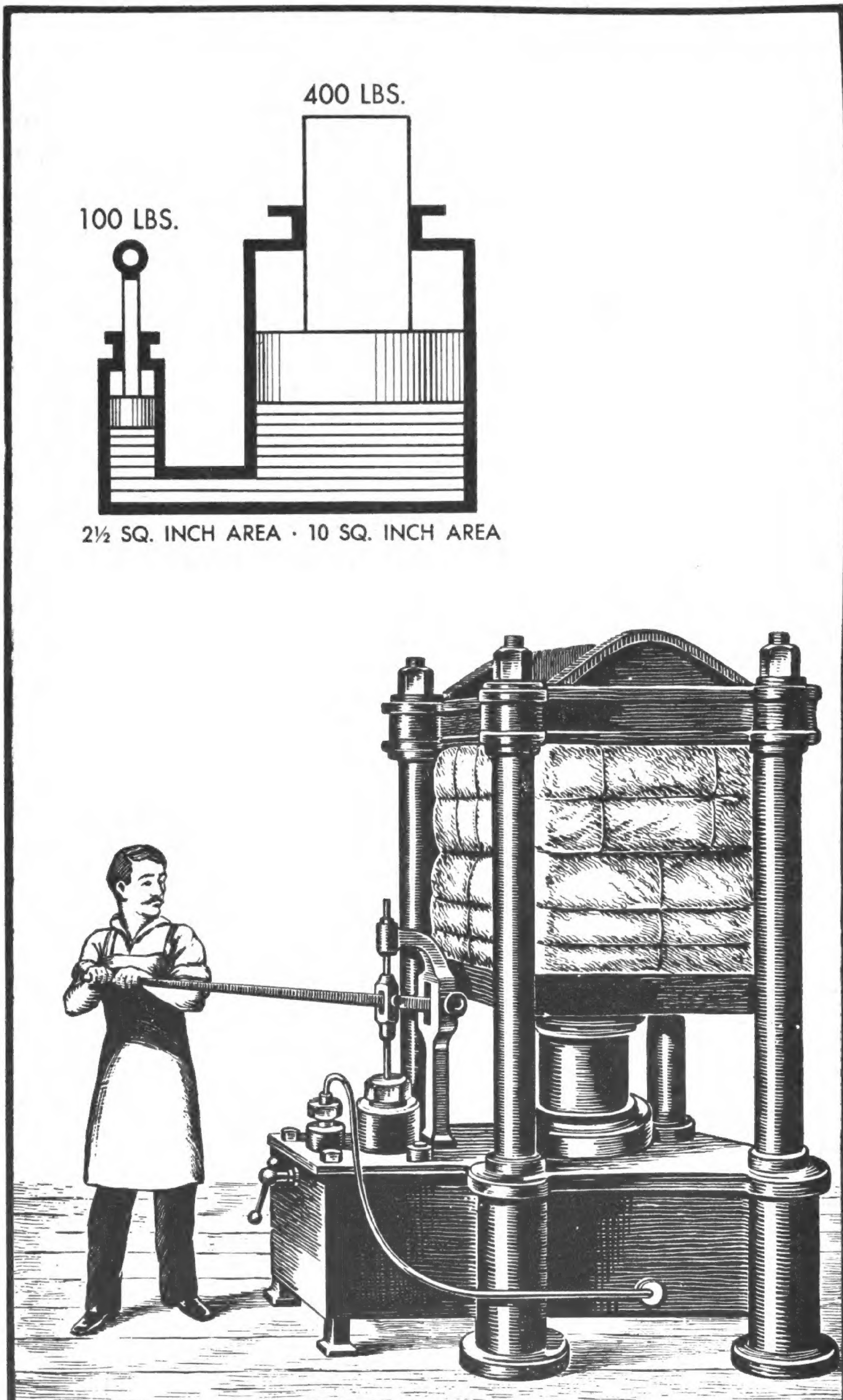
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Chapter 1

INTRODUCTION

This chapter discusses: the development of hydraulics; advantages and problems of hydraulic set-ups; the physical properties of liquids; the relations of pressure and force in hydraulic systems; two simple applications of hydraulic principles—the hydraulic jack and the four-wheel hydraulic brake system used on many automobiles; the influence of atmospheric pressure on hydraulic systems; equality of work input and output in hydraulic systems.

Development of Hydraulics

What hydraulics is. The word *hydraulics* is based on the Greek word for water, and originally covered a study of the physical behavior of water at rest and in motion. Use has broadened its meaning to cover the physical behavior of all liquids, including the oils so often used in present-day hydraulic systems.

The science includes, for example, the manner in which liquids act in tanks and pipes, dealing with their properties and with ways of utilizing these properties. It includes the laws of floating bodies and the behavior of liquids on the submerged surfaces of dams and sluice gates. It treats of the flow of liquids under various conditions, and of ways of directing this flow to useful ends, as well as many other related subjects.

This book will deal primarily with the phases of hydraulics relevant to mechanisms, and especially to mechanisms used in the U. S. Navy.

History. Although the modern development of hydraulics is of comparatively recent date, the ancients were familiar with many hydraulic principles and their applications. The Egyptians and the ancient peoples of Persia, India and China conveyed water along channels for irrigation and domestic purposes, using dams and sluice gates to control the flow. The ancient Cretans had an elaborate plumbing system. Archimedes studied the laws of floating and submerged bodies. The Romans constructed aqueducts to carry water to their cities.

After the breakup of the ancient world, there were few new developments for many centuries. Then over a comparatively short period, beginning not more than three or four hundred years ago, the physical sciences began to flourish, thanks to the discovery of principles basic to all of them, and to the invention of many new mechanical devices. Thus the fundamental law underlying the whole science of hydraulics was discovered by Pascal in the seventeenth century, and by the end of the next century ways had been found to make the snugly fitted parts required in hydraulic systems no less than in other modern machinery. Since then progress has been rapid.

Present day uses. Today, hydraulic tools and machines are to be found everywhere about us. In a garage, a mechanic raises the end of a car with a hydraulic jack. Many motorists carry a smaller model in their tool kits, although others use simple lever jacks. In service stations, hydraulic lifts are used to raise cars for chassis lubrication. Dentists and barbers call on hydraulics to lift their chairs to a convenient working height by a few strokes of a lever (Figure 1). Hydraulic elevators transfer merchandise in ware-

Figure 1

houses from one floor to another. Hydraulic door stops keep heavy doors from slamming (Figure 2). Fluid drive is used in some automobiles. City water systems depend upon the large scale application of hydraulic principles.

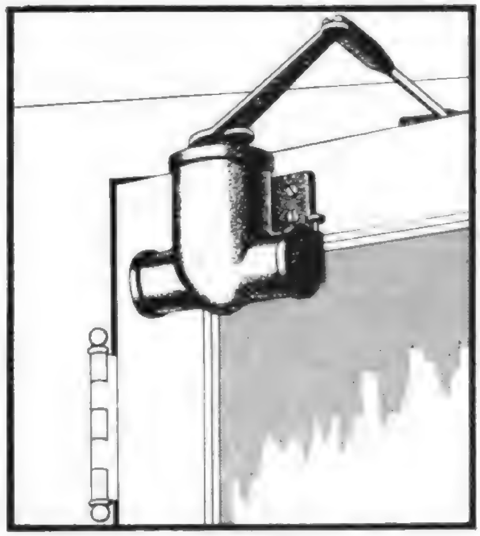


Figure 2

During the period preceding World War I, the Navy began to apply hydraulics extensively to naval mechanisms. Since then, naval applications have increased to the point where many ingenious hydraulic devices are used in the solution of problems of gunnery, aeronautics, and navigation. Aboard ship today, we see applications of hydraulics applied to anchor windlasses, power cranes, steering gear, remote control, power drives for the elevating of guns and training of mounts and turrets, for powder and projectile hoists, recoil systems, gun rammers, airplane catapults, and in many other places.

Advantages and Problems of Hydraulic Set-Ups

Advantages. The common use of hydraulic apparatus is due to the fact that properly constructed hydraulic systems possess a number of favorable characteristics. Although complex installations have been developed for special purposes, those in common use are generally simple in construction. They eliminate the need for complicated systems of gears, cams and levers. Motion can be transmitted without the slack inherent in the use of all solid machine parts. The liquids used are not subject to breakage as are mechanical parts, and the mechanism itself is not subject to great wear.

The different parts of a hydraulic set-up can be conveniently located at widely separated points, and the forces generated rapidly transmitted over considerable distances with small loss. These forces can be carried up and down or around corners, with small losses in efficiency and no complicated mechanisms. Very large forces can be

controlled by much smaller ones, and can be transmitted through comparatively small pipes and orifices.

If the system is well adapted to the work it is required to do, and if it is not misused, it can provide smooth, flexible, uniform action without vibration, unaffected by variations of load. An automatic release of pressure in case of overload can be guaranteed, so that the system is protected against breakdown or strain. Hydraulic systems can provide widely variable motions in both rotary and straight-line transmission of power. Need for control by hand can be minimized. Hydraulic systems, finally, are economical to operate.

Special problems. The extreme flexibility of hydraulic elements gives rise to a number of problems. Since fluids have no shape of their own, they must be positively confined throughout the entire system, and prevented from going anywhere except where we want them to go. We must give special thought to the structural organization and the relation of parts of a hydraulic system; we must provide strong pipes and containers; we must prevent leaks. This problem is acute with the high pressures obtained in many hydraulic installations.

The pressures set up in hydraulic systems must be controlled, as likewise the movement of the component fluids. This movement causes friction, within the liquid itself and against containing surfaces, which if excessive can lead to serious losses in efficiency. Dirt must not be allowed to accumulate in the system, where it will clog small passages or score nicely fitted parts. Chemical action may cause corrosion. Valves and pumps raise a variety of problems. Above all, it is necessary to know how a hydraulic system works, both in terms of the general principles common to all physical mechanisms and the peculiarities of the particular set-up at hand.

All physical science based on same principles. Although solids differ in certain ways from liquids and gases, and although every substance —lead, tin, water, oil, oxygen—has its own special properties, all kinds of matter shows the same fundamental patterns of action and reaction. When the principles common to all physical science became clarified, in early modern times, new possibilities of understanding

and action were opened up for mankind. So also today, when a person thoroughly grasps these principles, he is in a position to enlarge his power to understand and control the natural world.

Thus, although solids are rigid and fluids are not, the student can apply what he knows about levers to hydraulic situations. Just as a child weighing say 50 pounds can balance one weighing 100 pounds by sitting twice as far away from the balancing point, so in hydraulics a force of 50 pounds applied over a certain area can balance a force of 100 pounds acting over twice that area (Figure 3).

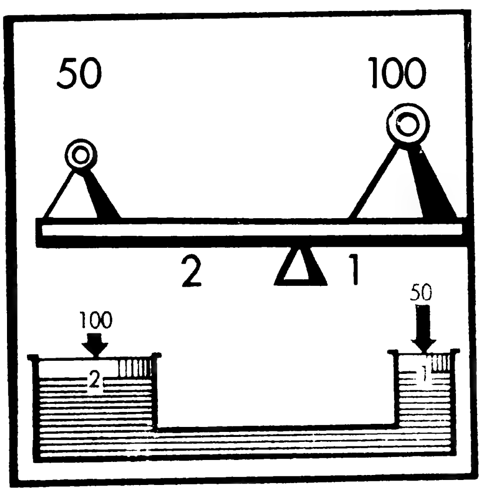


Figure 3

Different ways of doing the same thing. The inventor or engineer who wants to move something is therefore always faced with a number of possibilities. The thing to be moved could be pushed with a stick or pulled by a chain. A jet of water or a stream of air might play upon it, or something could be thrown against it. If a weight is to be lifted he might use levers, gears, ropes and pulleys, chain hoists, screw jacks, hydraulic jacks, or the force of flotation. Part of the system could make use of electricity, heat or even light, as in the case of doors that open automatically when people approach.

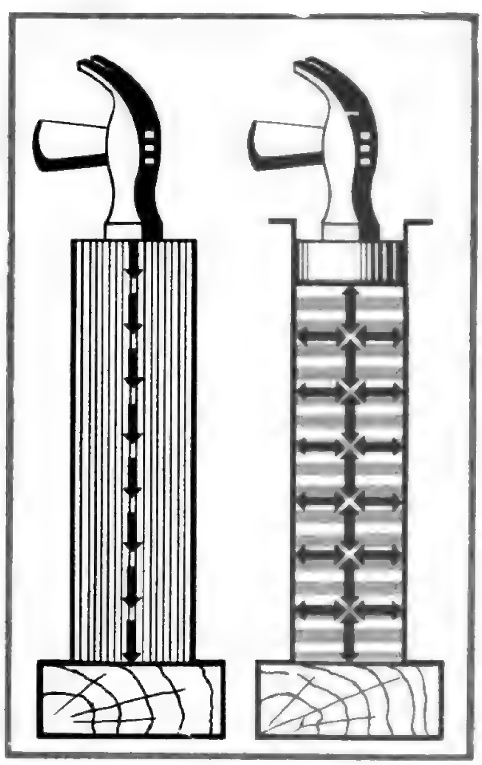
Each method of accomplishing mechanical movements has its good points and its bad, its special capabilities and limitations. Ropes and chains, for example, can pull but they cannot push; hydraulic elements can push but they cannot pull, solid rods can both push and pull but cannot go around corners like chains or ropes or hydraulic elements; gears can operate continuously in a rotary motion, whereas a lever usually operates so that the arms of the lever move back and forth. And so it goes for every machine element. Though each obeys the same general physical laws, each also possesses special properties that must be taken into account.

Physical Properties of Liquids

Shapelessness of liquids. While solids always have a definite shape, liquids have no outer form of their own. They quickly conform in shape to their containers. Because of their own shapelessness, we can lead liquids almost anywhere in a pipe or a hose by means of gravity or by applying forces to them. In ease of transmission they are second only to electricity, which can go anywhere without gravitational effect, and which requires only a conductor.

Incompressibility of liquids. In spite of having no shape of their own, liquids are even less compressible than most solids. When a force is applied to a confined liquid, the liquid exhibits substantially the same effect of rigidity as a solid. If an appropriate exit is provided, this effect can be combined with fluidity to transmit a force.

A force of 15 pounds on a cubic inch of water will decrease its volume by only 1/20,000. It would take a force of over 32 tons to reduce it 10 per cent. Other liquids do not behave differently. And when pressure on a liquid is removed, the liquid immediately returns to its former volume.



Figures 4 and 5

Transmission of forces through liquids. When we strike the end of a bar, the main force of the flow is carried straight through the bar to the other end (Figure 4). This happens because the bar is rigid. The direction of the blow almost entirely determines the direction of the transmitted force. The more rigid the bar, the less force is lost inside the bar or transmitted outward at right angles to the direction of the blow.

When we apply a force to the end of a column of confined liquid, however, it is transmitted not only straight through to the other end, but also equally and undiminished in every direction throughout the column—forwards, backwards and sideways—so that the containing vessel is literally filled with pressure (Figure 5)

For this reason a flat hose takes on a circular cross section when it is filled with water under pressure. The outward push of the water is equal in every direction (Figure 6). Water would leave the hose at the same velocity through leaks, no matter on what side of the hose they might happen to be. The velocity right at the point of exit would not be appreciably less in a leak on the upper side of the hose, nor more in one on the bottom.

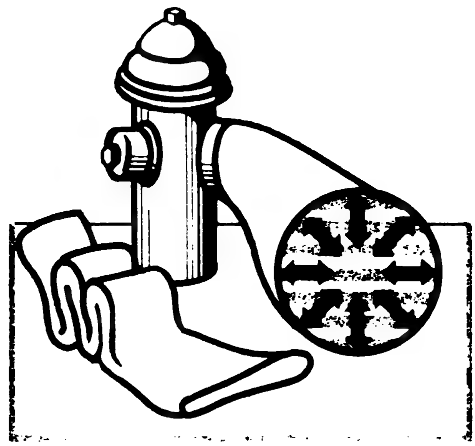


Figure 6

Pascal's Law. The foundations of modern hydraulics were laid in 1653 when Pascal discovered that *pressure set up in a liquid acts equally in all directions. This pressure acts at right angles to containing surfaces.* Thus in Figure 7, if the liquid standing on a square inch *A* at the bottom of the tank weighs 8 pounds, a pressure of 8 pounds per square inch will be exerted in every direction at *A*. The liquid resting on *A* will push equally downward and outward. But the liquid on every other square inch of the bottom surface is also pushing downward and outward in the same way, so that the pressures on different areas are in balance. At the edge of the bottom the pressures act against the wall of the tank, which everywhere must be strong enough to resist them with a force exactly equal to the push. Every square inch of the bottom of the tank also is strong enough to resist the downward pressure of the liquid resting on it. The same balance of pressures exists at every other level in the tank, though of course at lesser pressures as one approaches the surface. Therefore, the liquid remains at rest: it does not leak out and the tank does not collapse.

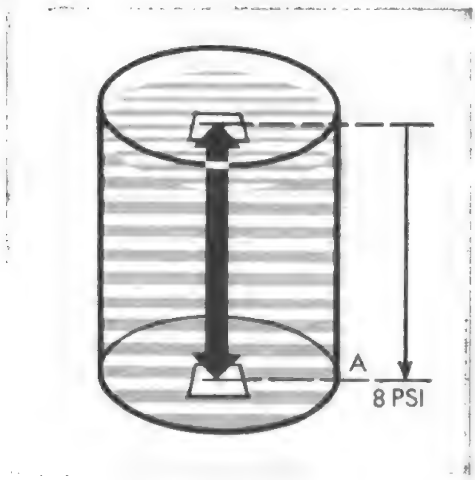


Figure 7

One of the consequences of Pascal's Law is that the shape of the container in no way alters pressure relations. Thus in Figure 8, if the pressure due to the weight of the liquid at one point on the horizontal line *H* is 8 pounds per square inch, the pressure will be 8 pounds per square inch everywhere at level *H* in the system.

Pressure due to a liquid's weight will depend, at any level, upon the vertical height of liquid from that level to the surface of the liquid. *The vertical distance between two horizontal levels in a liquid is known as the head of the liquid* at the lower level with respect to the higher. In Figure 8, the liquid head of all points on level *H* with respect to the surface is indicated.

Density and specific gravity. Pressure due to liquid head will also depend upon the density of the liquid—its weight, that is, per unit of volume. Water, for example, weighs 62.4 pounds per cubic foot or 0.036 pound per cubic inch, while a certain oil might weigh 55 pounds per cubic foot, or 0.032 pound per cubic inch. It would take 222 inches of head using water to produce a pressure of 8 pounds per square inch, and 252 inches using the oil (Figure 9).

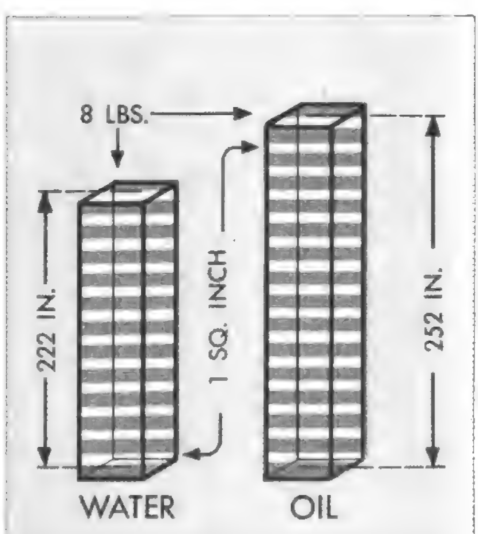


Figure 9

The specific gravity of a substance is the ratio of the weight of a unit volume of that substance—its density—to the weight of the same volume of some standard substance, measured under standard pressure and

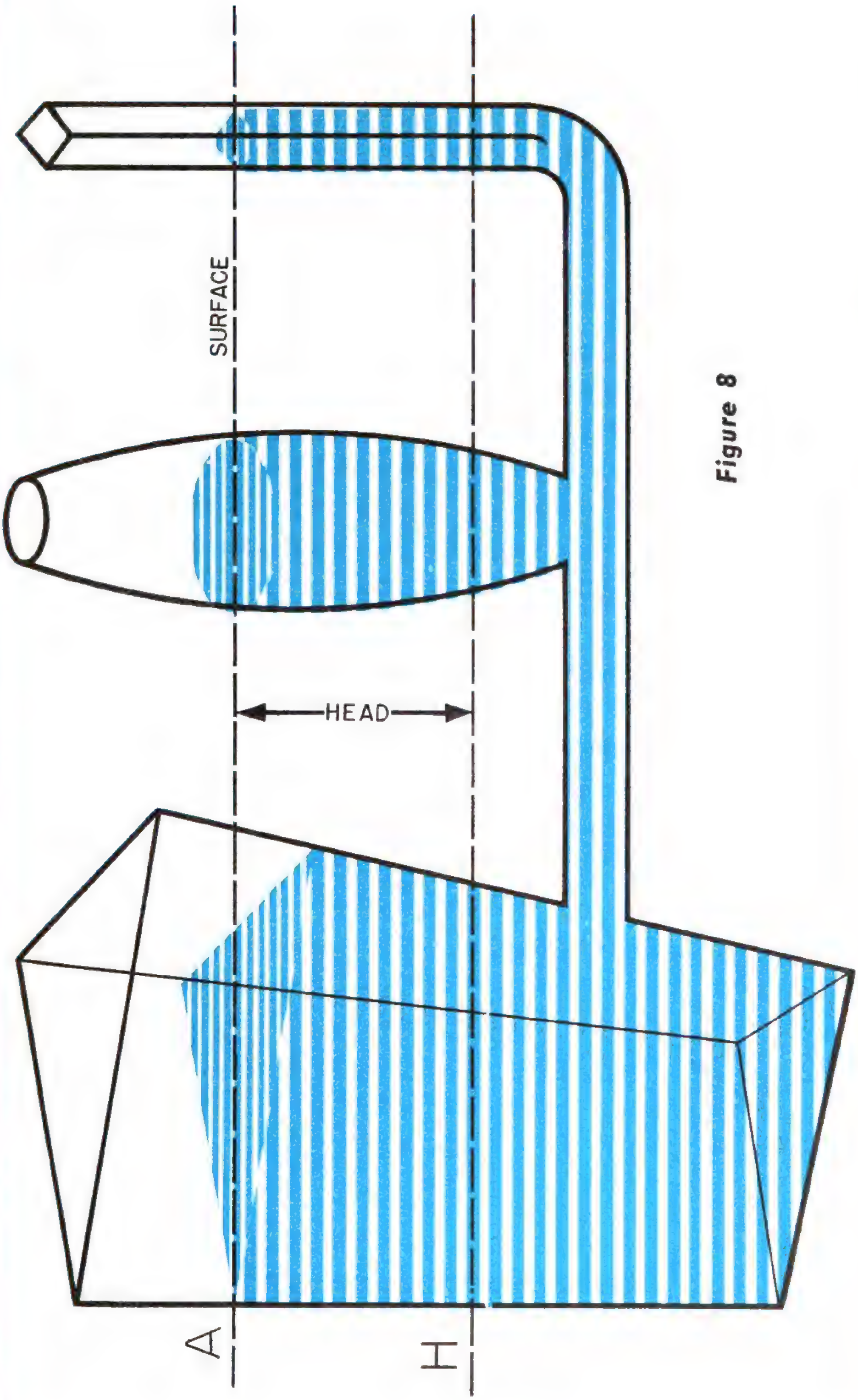


Figure 8

temperature conditions. Since water is the standard used for liquids and solids, the specific gravity of the oil we have mentioned is 55/62.4 or 0.881. Volume for volume under standard conditions, in other words, this oil has 881-thousandths the weight of water.

Pressure and Force in Hydraulic Systems

Application and transmission of forces. We have just seen that, in accordance with Pascal's Law, any force applied to a confined liquid is transmitted equally in all directions throughout the liquid regardless of the shape of the container. Let us now consider the effect of this in the system shown in Figure 10. This is in reality a modification of Figure 5 in which the column of liquid is curved back upward to its original level, with a second piston at this point. It is clear that if we push down on piston *1* a pressure will be created throughout the liquid, which will act equally at right angles to surfaces in all parts of the container.

Pressure is defined as force divided by the area over which it is distributed. In the case of Figure 10, if the force *1* is 100 pounds and the area of the piston 10 square inches, then the pressure in the liquid must be 10 pounds per square inch. This is sometimes written 10 psi.

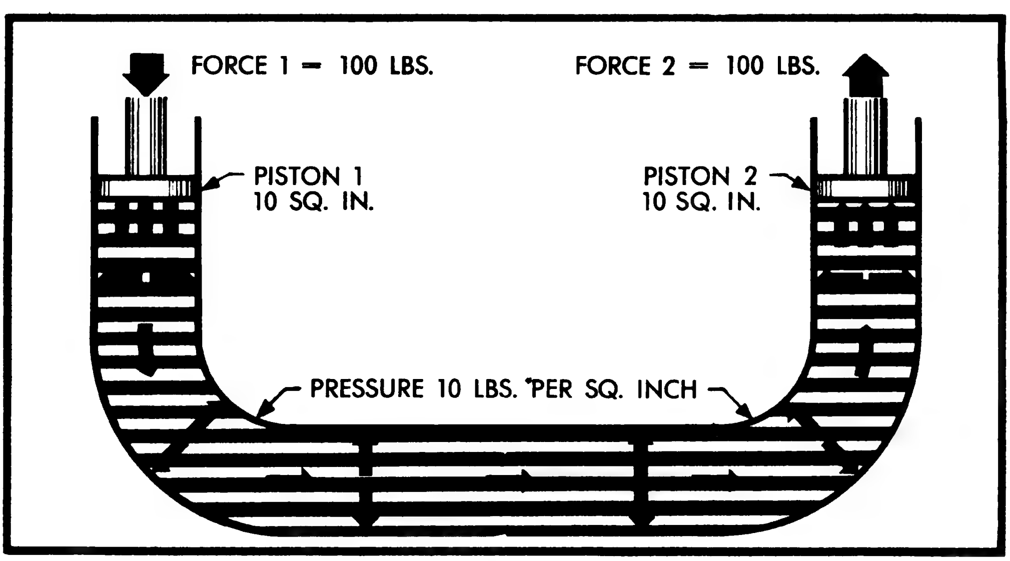


Figure 10

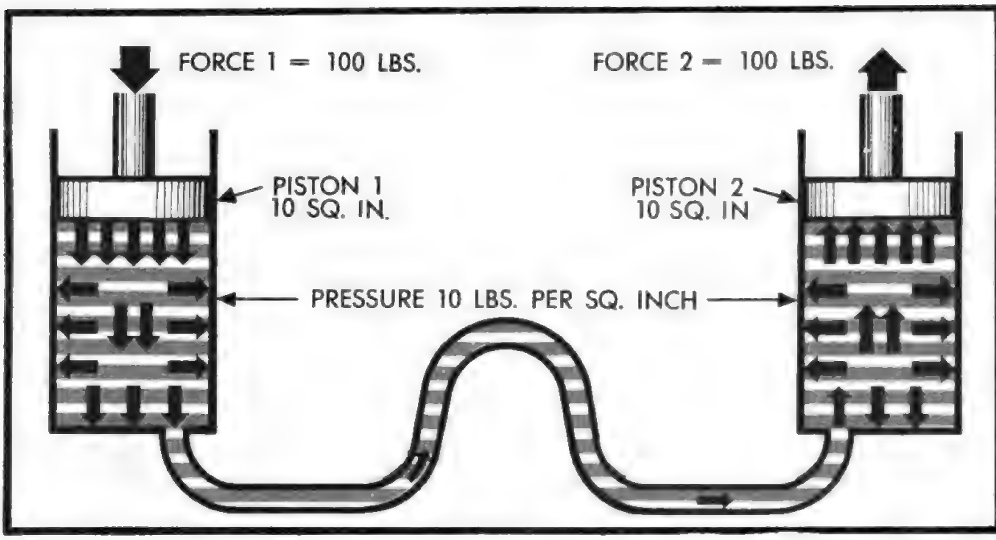


Figure 11

This pressure must act also on piston *2*, so that for every square inch of its area it will be pushed upward with a force of 10 pounds. In this case we are considering a liquid column of uniform cross section so that the area of piston *2* is the same as piston *1*, or 10 square inches. Therefore, the upward force *2* on piston *2* will be 100 pounds, the same as was applied to piston *1*. All we have done in this case is to carry our 100-pound force around a bend, but the principle illustrated underlies practically all mechanical applications of hydraulics.

At this point it should be noted that since Pascal's Law is independent of the shape of the container, it is not necessary that the tube connecting the two pistons should be the full area of the pistons throughout. A connection of any size, shape, or length would do, so long as an unobstructed passage is provided. Therefore the system shown in Figure 11, wherein a relatively small bent pipe connects two cylinders, would act exactly the same as that shown in Figure 10.

Multiplication of forces. In Figures 10 and 11 we considered hydraulic systems having pistons of equal area wherein the output force was equal to the input force. Let us now consider the situation of Figure 12, where the input piston is much smaller and the output piston much larger. Let us assume that the area of piston *1*

is now 2 square inches and that of piston *2* is 20 square inches. Now if we push down on piston *1* with a force of 20 pounds, the pressure created in the liquid will again be 10 pounds per square inch, although the original force is much smaller. This is because the force is concentrated on a relatively smaller area.

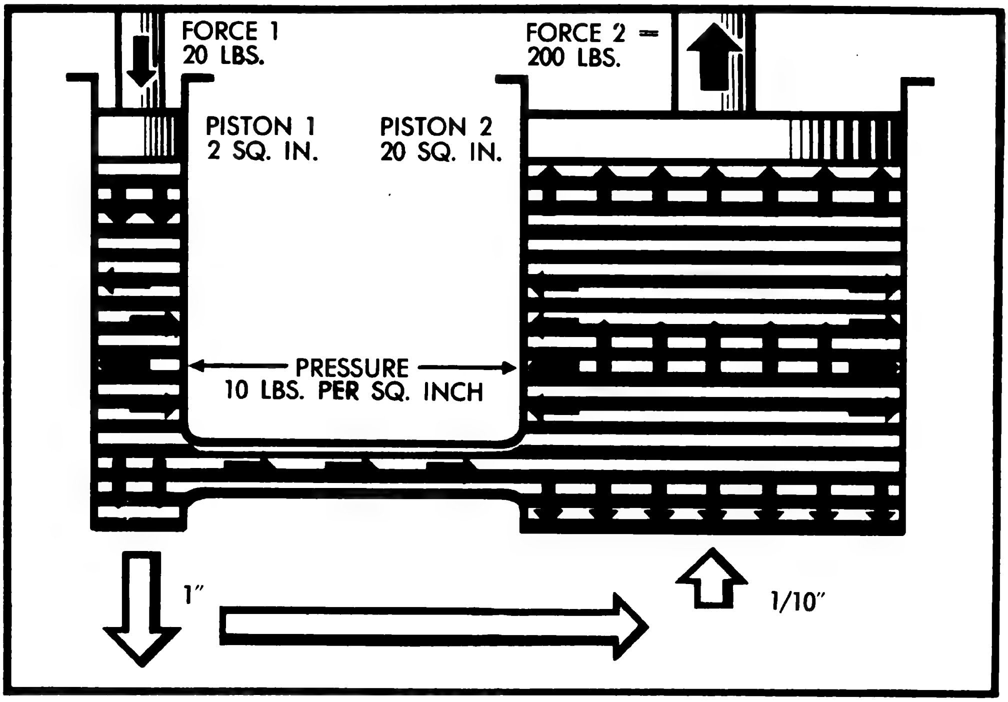


Figure 12

As we have already seen, this pressure of 10 pounds per square inch will act on all parts of the liquid container, including the bottom of piston *2*. The upward force on piston *2* will therefore be 10 pounds for each of its 20 square inches of area, or 200 pounds. In this case we have multiplied our original force tenfold while using the same pressure in the liquid as before. Obviously, the system would work just the same for any other forces and pressures, so that the ratio of output force to input force will always be ten to one.

The system will of course work the same in reverse. If we consider *2* the input piston and *1* the output piston, then the output force will always be one-tenth the input force. Sometimes we wish to secure just such a result.

We can now state the general rule that, because of Pascal's Law,

if two pistons are used in a hydraulic system, *the force acting on each will be directly proportional to its area, and the magnitude of each force will be the product of the pressure and its area.*

Differential areas on a piston. Let us now consider the special situation shown in Figure 13. Here we have a single piston *1* in a cylinder *2* having a piston rod *3* attached to one side of the piston and extending out of the cylinder at one end. Liquid under pressure is admitted to both ends of the cylinder equally through the pipes, *4*, *5* and *6*. The opposed faces of piston *1* behave like two pistons acting against each other. The area of one face is the full area of the cylinder, say 6 square inches, while the area of the other face is the area of the cylinder minus the area of the piston rod, which is 2 square inches, leaving an effective area of 4 square

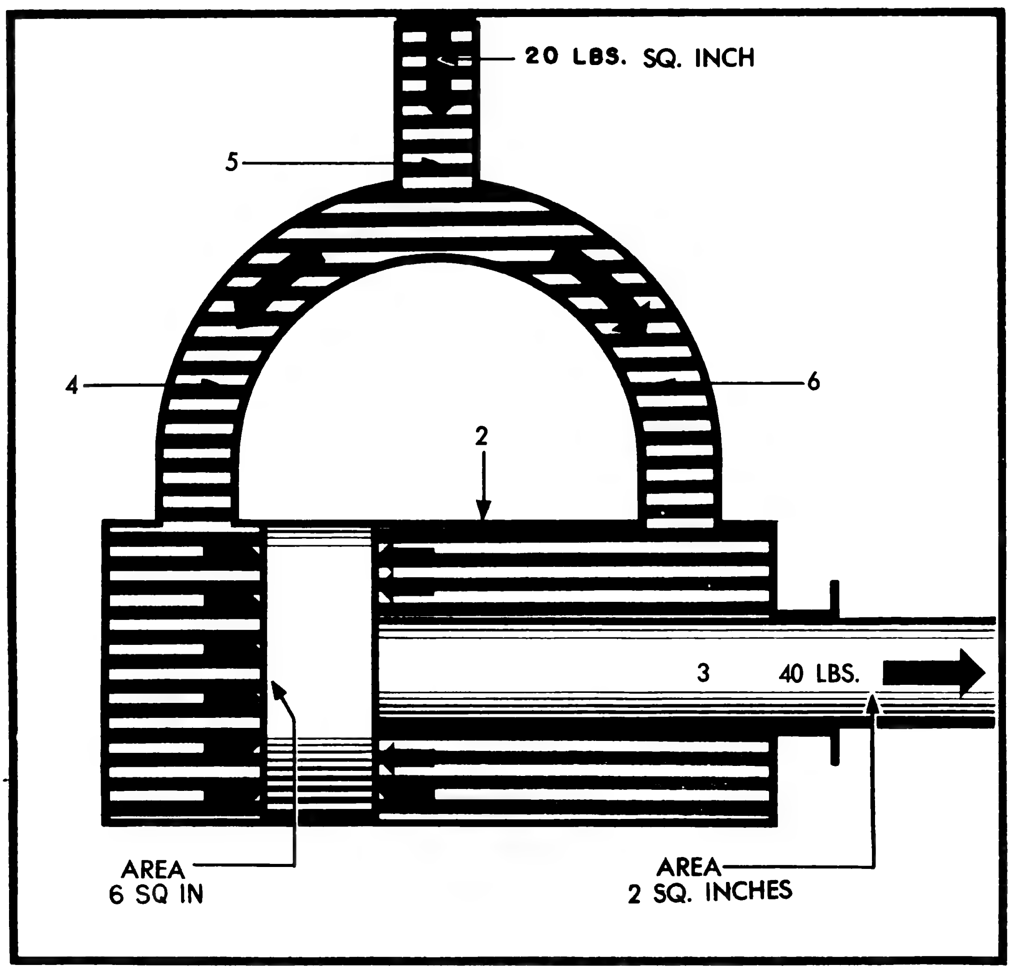


Figure 13

inches on the right face of the piston. The pressure on both faces will of course be the same, say 20 pounds per square inch. Applying the rule just stated, we find that the force pushing the piston to the right will be its area times the pressure, $20 \times 6 = 120$ pounds. Likewise the force pushing it to the left will be its area times the pressure, or 80 pounds. Therefore, there is a net unbalanced force of 40 pounds acting to the right, and the piston will move in that direction. The net effect is the same as if the piston and cylinder were just the size of the piston rod, since all other forces are in balance.

Volume and distance factors. In the systems of Figures 10 and 11 the pistons had areas of 10 square inches each. Therefore if we pushed one of them down 1 inch, 10 cubic inches of liquid would be displaced. Since the liquid is practically incompressible, this volume of liquid must go some place. The only thing for it to do is to move the other piston. Since the area of this piston is likewise 10 square inches it will have to move 1 inch in order to accommodate the 10 cubic inches of liquid. The pistons are of equal areas, and will therefore move equal distances, though in opposite directions.

Applying this reasoning to the system of Figure 12, we find that if piston *1* is pushed down 1 inch only 2 cubic inches of liquid will be displaced. In order to accommodate these 2 cubic inches of liquid piston 2 will have to move only 1/10 of an inch, because its area is 10 times that of *1*. This leads to the second basic rule for two pistons in the same hydraulic system, which is that *the distances moved are inversely proportional to their areas.*

Hydraulic Jacks and Brakes

General set-up for hydraulic jack. Suppose we need to lift heavy weights, and that only a small force—for example hand power—is available for the purpose. The use of a hydraulic jack will solve the problem.

In Figure 14 the small piston has an area of 5 square inches. The small cylinder is directly connected to a large cylinder with a pis-

ton having an area of 250 square inches. The top of this piston forms a lift platform.

Now if a force of 25 pounds is applied to the small piston, it produces a pressure of 5 pounds per square inch in the liquid. This pressure acting on the 250 square inch area of the output piston will make a force of 1250 pounds available to raise the lift. An input force of 25 pounds has been transformed into a working force of more than half a ton.

With respect to distances traversed, however, the story is the exact opposite. Lowering of the small piston 5 inches will displace 25 cubic inches of liquid, and this distributed over the 250 square inch area of the large cylinder will raise its piston only 25/250 = 1/10 inch.

Practical problems. We have managed to raise 1250 pounds 1/10 inch. But it is most unlikely that anyone would ever want to stop at this. Unless the weight can be raised to a greater height, in most cases no practical result will be achieved. Our machine as it stands

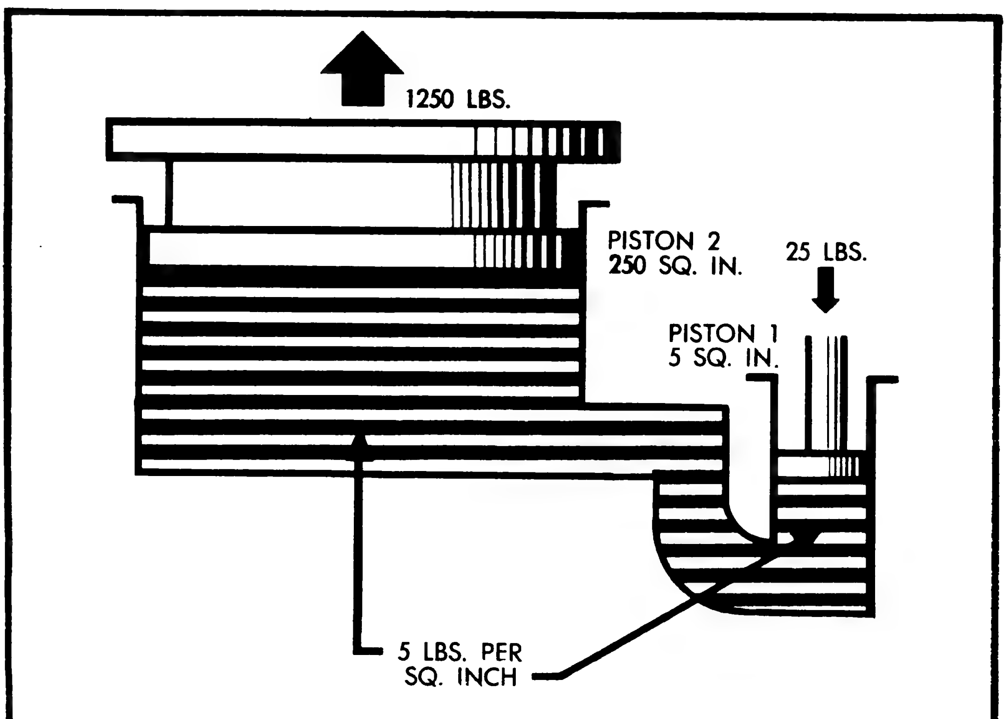


Figure 14

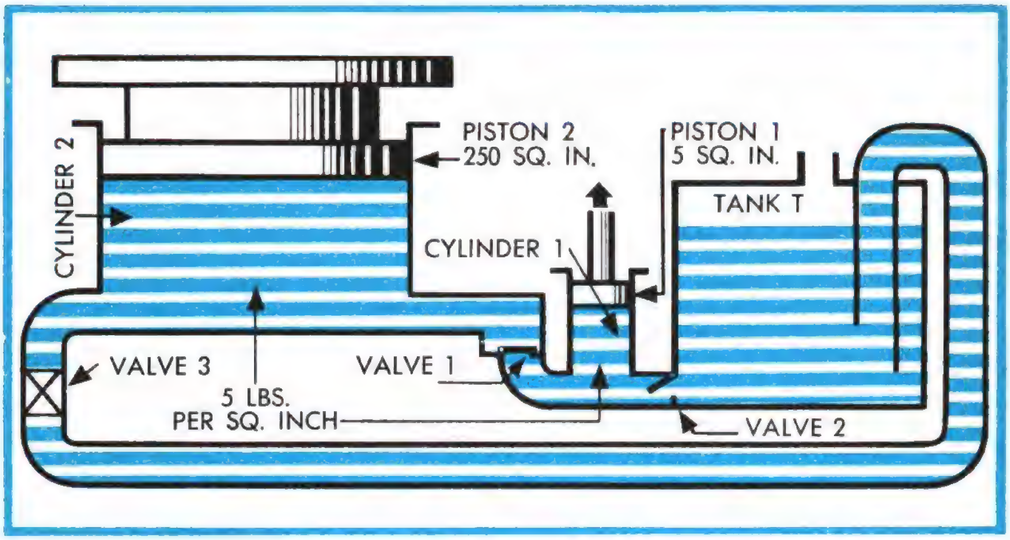


Figure 15

is not even fitted to maintain the advantage it has gained. If the force on piston *1* is released, the lift piston will drop back to its original position, and things will stand as they were at the beginning.

We must do something to hold the lift platform in place at the new level, after it has been raised, and we must provide some means of repeatedly raising and lowering piston *1*, gaining 1/10 inch at each double stroke. We must also find some way of lowering the lift

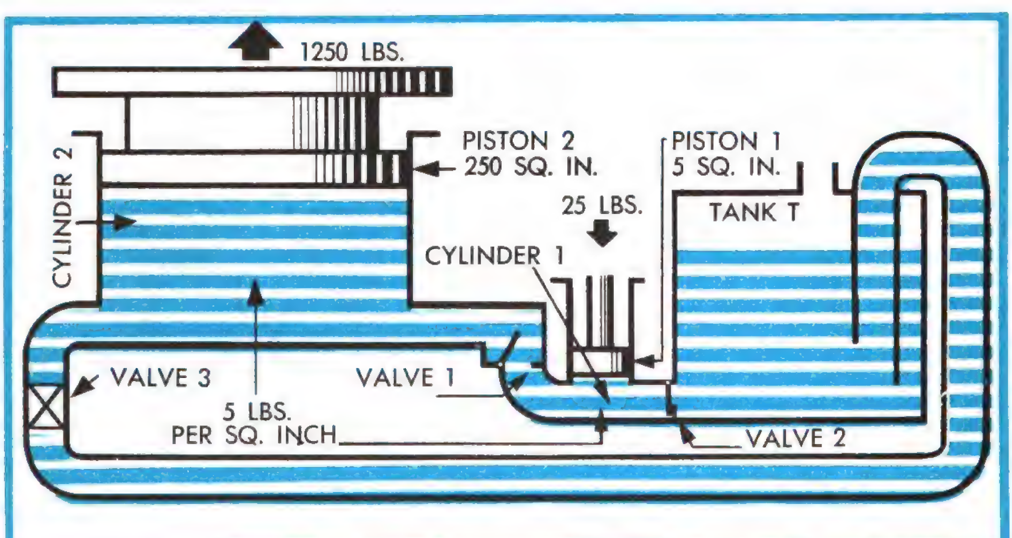


Figure 16

platform to any desired level. Then we shall have a hydraulic jack capable of raising 1250 pounds a considerable distance by hand, although at the rate of only 1/10 inch per stroke.

These results are attained by introducing a number of valves, and also a reserve supply of liquid to be used in the system (Figures 15 and 16). As the input piston *1* is raised (Figure 15), valve *1* will be closed by the back pressure from the weight on piston *2*, but valve *2* will be opened by the head of liquid in reserve tank *T*. This will force liquid into cylinder *1*. When piston *1* is lowered

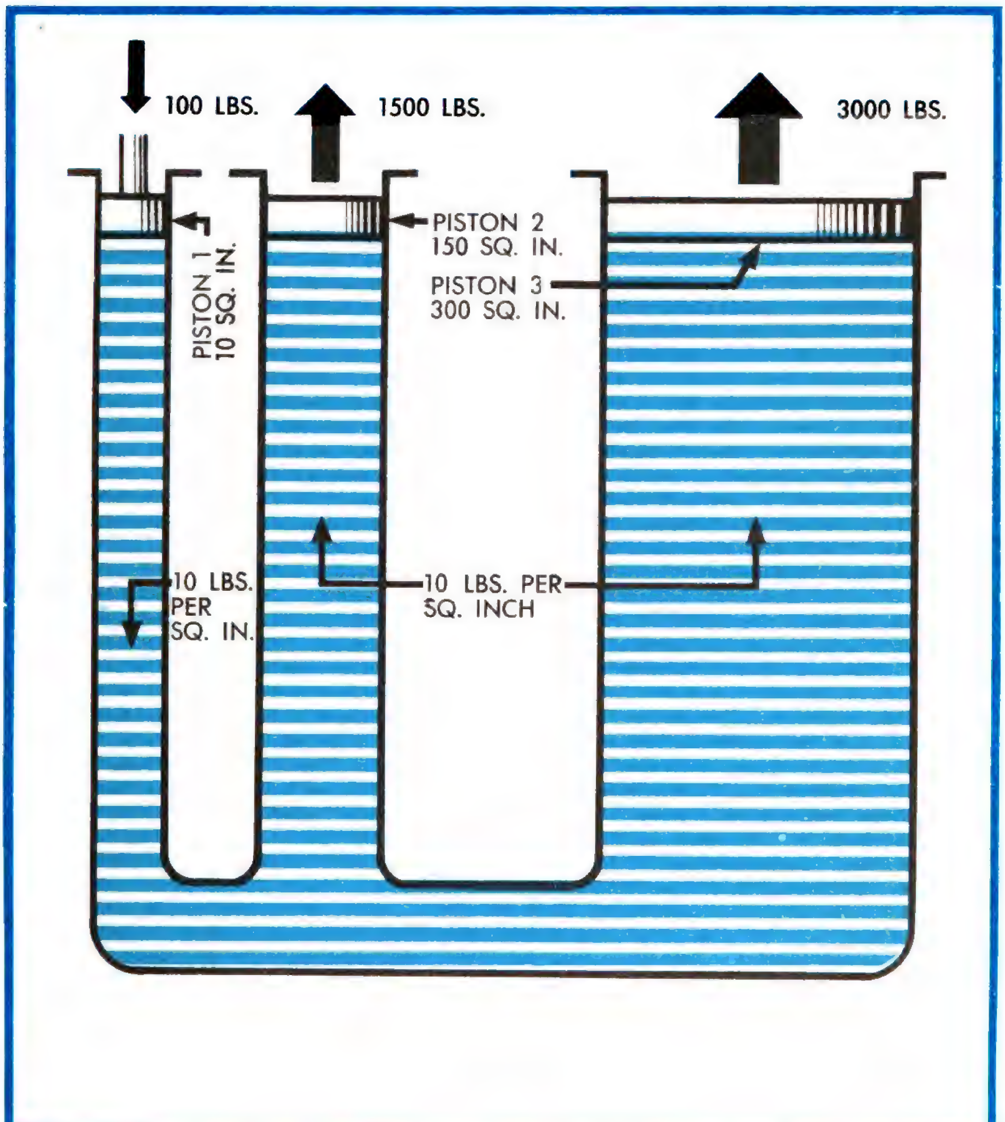


Figure 17

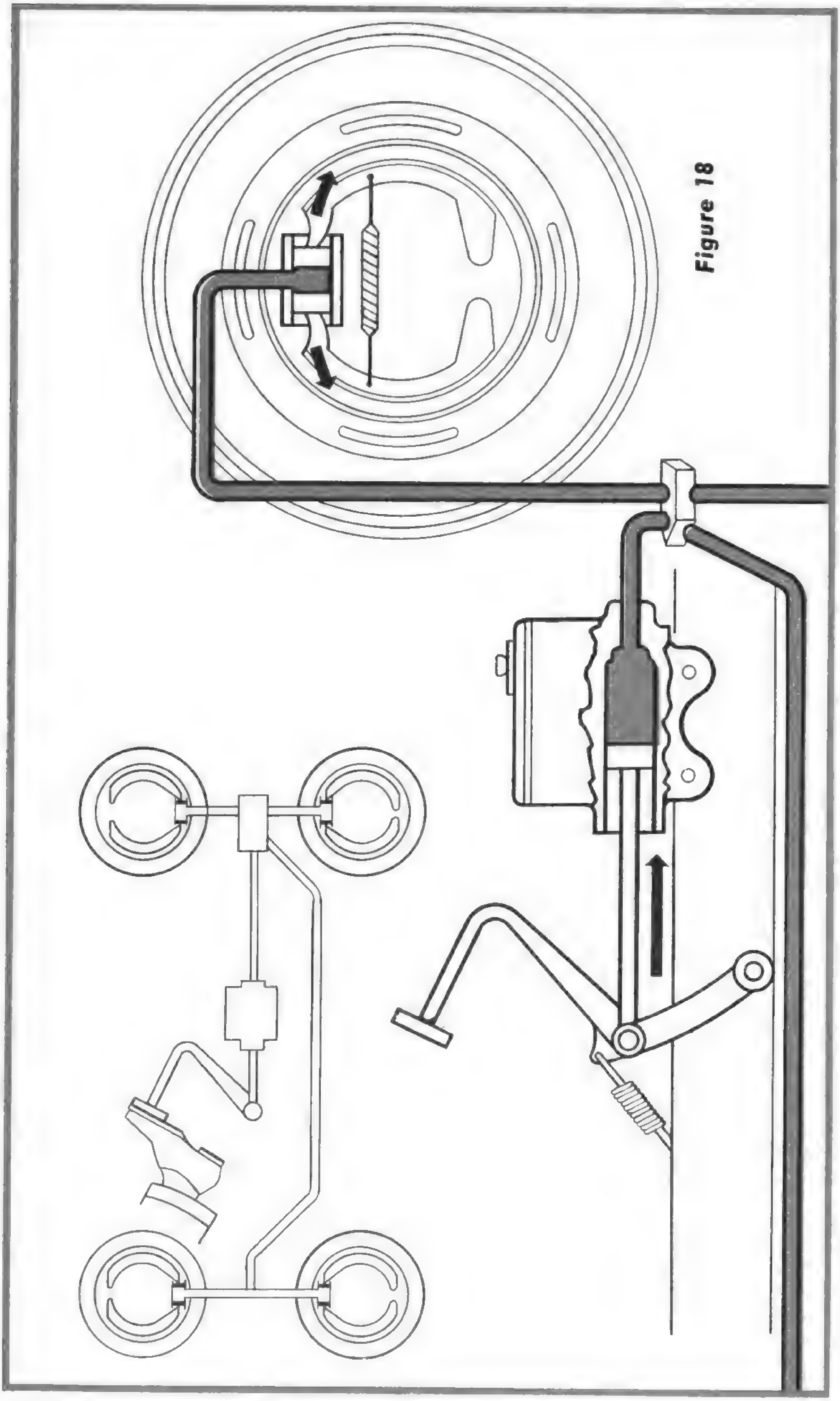


Figure 18

(Figure 16), a pressure will be developed in cylinder *1*. When this exceeds the head in tank *T*, it will close valve *2* and when it exceeds the back pressure from piston *2* it will open valve *1*, forcing liquid into the pipe line. The pressure from cylinder *1* is thus transmitted into cylinder *2*, where it acts to raise piston *2* with its attached platform. When piston *1* is again raised (Figure 15), so that the pressure in cylinder *1* falls below that in cylinder *2*, valve *1* will close, preventing return of the liquid and holding piston *2* at its new level. Thus by repeated strokes of piston *1* we can progressively raise piston *2* and its load as desired. To lower the lift platform, valve *3* is opened, and the liquid from cylinder *2* is returned to tank *T*.

The student should carefully verify the fact that the working hydraulic jack is built in conformity with the ideas he has been studying. He should note how, when a force is applied to a relatively incompressible liquid an equal pressure is set up throughout the liquid, according to Pascal's Law, establishing forces on surfaces directly proportional to their areas, and where these surfaces are free to move, carrying them through distances inversely proportional to their areas.

Multiple piston hydraulic jack. Forces can be transmitted by pressure to two or more lifts in the manner indicated in the diagram of Figure 17. Note that the pressure set up by the force applied to piston *1* is transmitted undiminished to both piston *2* and piston *3*, and that the resultant force on each piston is proportional to its area. The multiplication of output pistons introduces no new principle.

Figure 17 is only a general diagram. In order to show a workable machine it would have to be revised after the manner of Figures 15 and 16.

Hydraulic brakes. A practical application of multiple outlet pistons to distribute forces can be found in the four-wheel hydraulic brake system used on many automobiles (Figure 18). The foot pedal used by the operator is attached to a master input piston. The cylinder in which it moves is connected by tubing to a cylinder at each wheel which contains two opposed output pistons, each of which

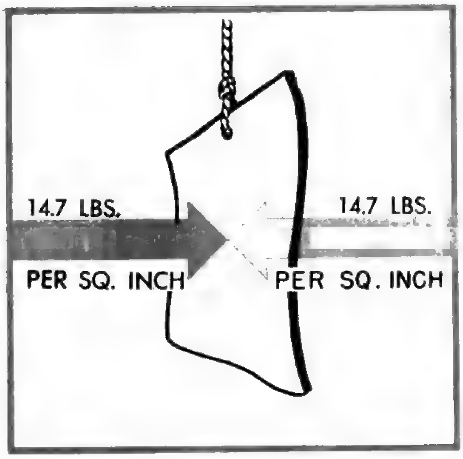


Figure 19

is attached to a brake shoe fitted inside the brake drum. When force is applied to the foot pedal, pressure is transmitted equally throughout the liquid to all eight output pistons, and each piston pushes a brake band against the wall of its brake drum, thus retarding the rotation of the wheel. When the operator lifts his foot from the pedal, pressure in the system is removed, and coiled springs return the brake shoes and the output pistons to their original positions.

In this manner, a force applied to the brake pedal will produce a proportional force on each of the output pistons, which will in turn apply the brake bands frictionally to the turning wheels to retard rotation. If all eight output pistons were of the same size, the force exerted on each brake band would be exactly the same, provided frictional losses from the master cylinder to each wheel were equal. If the brake bands also were identical each wheel would be equally retarded.

In actual brake designs, however, it is customary to use a greater piston area for the rear wheels than for the front. This produces a greater braking action at the rear in a constant and predetermined ratio, and helps to keep the car from slewing around when stopping suddenly on slippery pavements.

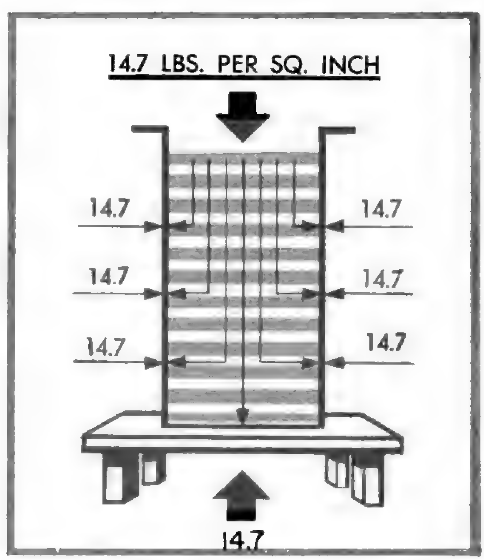


Figure 20

Atmospheric Pressure

Acts like other forces. Up to this point we have ignored the

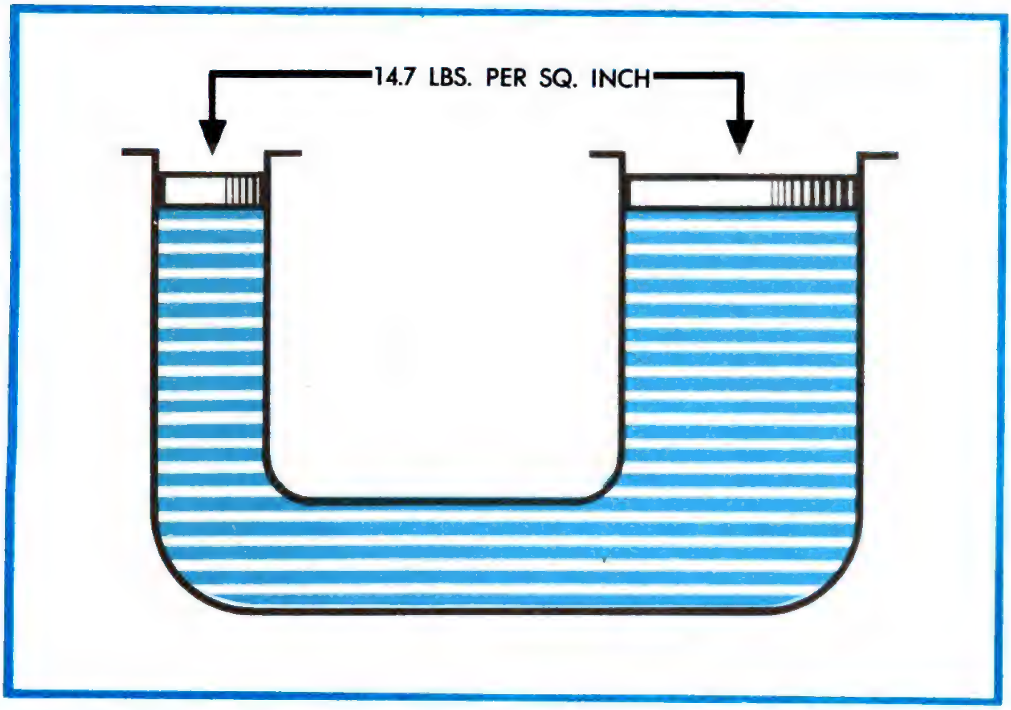


Figure 21

fact that all of the hydraulic systems we have been discussing are literally bathed in air. Everything on the surface of the earth exists at the bottom of an ocean of air extending upward many miles into space. Even when a plane is flying at a great height it has ascended only a small way towards the point where the density of our atmosphere approaches zero.

Since the air in which we live is a mixture of gases — that is to say of material substances — it has weight and therefore exerts pressure by virtue of head. Although any small volume of air weighs very little, the pressure at sea level under standard conditions due to atmospheric head amounts to 14.7 pounds per square inch.

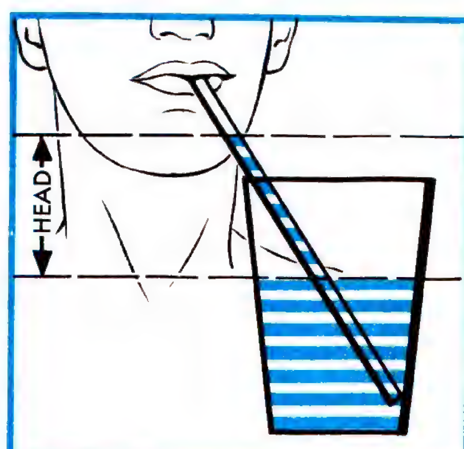


Figure 22

Since gases are compressed by their own weight, the same volume of air at sea level will

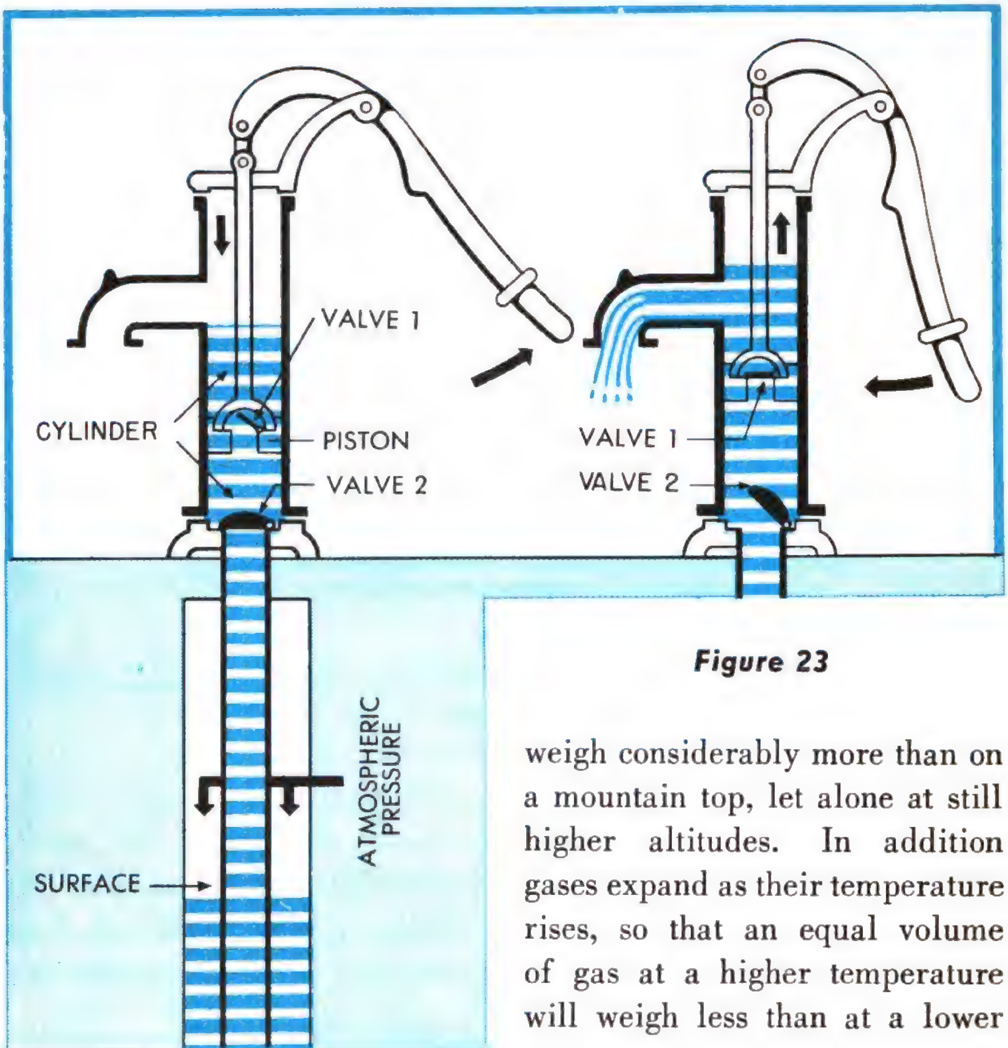


Figure 23

weigh considerably more than on a mountain top, let alone at still higher altitudes. In addition gases expand as their temperature rises, so that an equal volume of gas at a higher temperature will weigh less than at a lower temperature.

How is it possible to ignore a pressure of 14.7 pounds on every square inch of our body surface, as well as on the hydraulic systems we have been studying? The answer is simple. Atmospheric pressure is balanced either by itself or by the resistance of bodies strong enough to withstand it. Whenever this is not the case, we have an unbalanced force, and accelerated motion must ensue until the force is balanced.

Atmospheric pressures, in other words, obey Pascal's Law no less than do pressures, set up in liquids. Just as in Figure 7 we showed that pressures due to liquid head must balance at every point in every direction in a liquid if the liquid is to remain at rest, so too with atmospheric pressures. Nor is the situation changed if these

pressures act on opposite sides of even the thinnest surface, or through liquids. In Figure 19 the suspended sheet of paper is not torn by atmospheric pressure, as it would be by an unbalanced force of 14.7 pounds per square inch, because atmospheric pressure acts equally on both sides of the paper. In Figure 20 atmospheric pressure acting on the surface of the liquid is transmitted equally throughout the liquid to the walls of the container, but is balanced by the same pressure acting directly on the outer walls of the container. In Figure 21 atmospheric pressure acting on the surface of one piston is balanced by the same pressure acting on the surface of the other. The different areas of the two surfaces make no difference, since for a unit of area pressures are in balance.

Only if the difference in level of the two surfaces in Figure 21 was very great would the difference in atmospheric pressure produce an appreciable head of liquid.

Vacuum and partial vacuum. When you suck soda through a straw you pull some of the air from the straw and disturb the balance of pressures prevailing in the glass. Unbalanced atmospheric pressure acting on the surface of the liquid pushes soda up into the straw. The motion of the liquid will continue until a new balance is reached. The soda can be held at a certain level in the straw. This level will always be where the pressure of the head of liquid exactly equals the difference between the pressure in the straw and that on the surface of the liquid (Figure 22). If you take the straw from your mouth, the soda in the straw is subject to the same pressure as the surface of the liquid, and it falls back into the glass.

You have produced *a partial vacuum* in the straw—a pressure, this is to say, *less than the prevailing atmospheric pressure.* The theoretical limit of this process would be a condition of zero pressure—a complete vacuum. In actual practice, however, it is impossible to produce a complete vacuum.

The hand pump. Water is raised from a well by means of a hand pump by producing a partial vacuum in the flow pipe (Figure 23). The pump handle is attached to a piston in which is mounted valve *1*,

which swings open as the piston is lowered. Farther down the pipe is valve *2*. When the piston is raised, valve *1* closes, and the water held above it is carried up the pipe with the piston. A partial vacuum is produced in the space between the bottom of the piston and valve *2*, in the cylinder, so that the pressures on the two sides of valve *2* are unbalanced. The greater force on the under side of the valve, due to atmospheric pressure transmitted through the liquid up the pipe, opens valve *2* and water enters the cylinder, above the valve, where it is trapped by the closing of valve *2* during downward motions of the piston. Thus water is forced into the space above valve *2* during upward motions of the piston, and is admitted into the space above the piston during its downward motions, to be carried to the exit spout on the next upward motion of the piston.

If a perfect vacuum could be produced, the head as between valve *2* and the surface of the liquid could be 34 feet at sea level, since a 34-foot head of water produces a pressure of 14.7 pounds per square inch. Under actual working conditions, however, about 25 feet is considered a satisfactory lift for a water pump. For an oil pump it would be greater, since the specific gravity of oils is less than that of water.

Input and Output Relations

Equality of work input and output. In our consideration of the hydraulic jack (Figure 16), we found that an increase in output force over input force was accompanied by a decrease in the distance moved in exactly the same ratio. An increase in force can be obtained only by a proportionate decrease in distance moved. This is also true if we operate in the reverse direction: a distance increase can be obtained, but only at the expense of a force decrease in the same ratio. This leads us to the basic statement that, *neglecting friction, in any hydraulic system (or any other mechanical system for that matter) the input force multiplied by the distance through which it moves is always exactly equal to the output force multiplied by the distance through which it moves.*

Work and energy. We must now explain two new terms—*work* and

energy. The scientist defines work as a force moving through some distance, and *the amount of work done is the product of the force multiplied by the distance through which it moves*. Therefore the rule just stated means that, when friction is neglected, the work output is always equal to the work input. Energy is a somewhat similar but broader term. *Energy* includes work and in addition *all the forms into which work can be converted or which can be converted into work*. Work always involves actual movement, but energy can be at rest and still exist as energy, so long as it is capable of doing work. Work and energy are always mutually interchangeable.

In our study ot the hydraulic jack we neglected the subject of friction for the sake of simplicity. However, it is well known that there is always some friction in actual machines. It is also known that heat is produced whenever work is done against friction. Therefore, heat is a form of energy because it can be produced from work. Likewise heat in the form of fire under a boiler can be converted into work through the medium of a steam engine. Now we know that in practice friction represents a loss of efficiency, but this does not mean an annihilation of energy itself. It means only that some of the energy put into the system has been converted into another form which is not useful for the particular problem in hand. The energy is not usable or available, but it still exists as dissipated heat.

In agreement with this fact, any work or energy added to the system must in turn come from somewhere else, for it is never possible to create or destroy energy. All that can be done is to change it from one form into other forms, so as to make it more or less applicable to the purposes in hand. In the case of an actual hydraulic jack, since there is always some friction both within the liquid and between adjacent parts, the *useful* work output will not exactly equal the work input, but the difference will always exist somewhere in some other form of energy. In this case it will appear as heat which must escape from the system somewhere at some time. In other words, while the usable work output does not equal the input, *the total energy output in all forms will always exactly equal the*

total energy input. This is known as *the law of the conservation of energy.*

Energy can exist in a great many different forms, but they all have one thing in common; they are all interchangeable with each other and with work. The many forms which energy can take and their interchangeability is illustrated by a hydro-electric plant. Here a body of water is held back by a dam (Figure 24). In this case the water represents *potential energy,* because it is not doing work at the moment but is capable of doing work if it is released. If an opening is provided, water will rush out in a high velocity jet representing energy of motion or *kinetic energy.* If this jet is played against the blades of a water wheel it will push them around producing a continuous rotary motion. This is work in its true sense because a force is moving through a distance. The water wheel can in turn be connected to an electric generator which converts the work into electricity. This electricity can in turn be converted back into work by the use of an electric motor. Or it can be converted into light by the use of an electric bulb or into heat in an electric iron. By means of a motor and a pump we could even transform the energy all the way back into its original form of potential energy existing as a body of water at an elevation. Thus all of these forms of energy are interchangeable with each other. In actual mechanisms there is always some loss in the form of friction producing heat at every exchange, but the total energy, useful and wasted, will always add up to the original energy input.

QUESTIONS

1. Give some examples of the application of hydraulic principles in ancient and modern times.

2. What are some of the special features and problems of hydraulic machines?

3. What is meant by the incompressibility of liquids?

4. How does the transmission of forces in liquids differ from their transmission in solids?

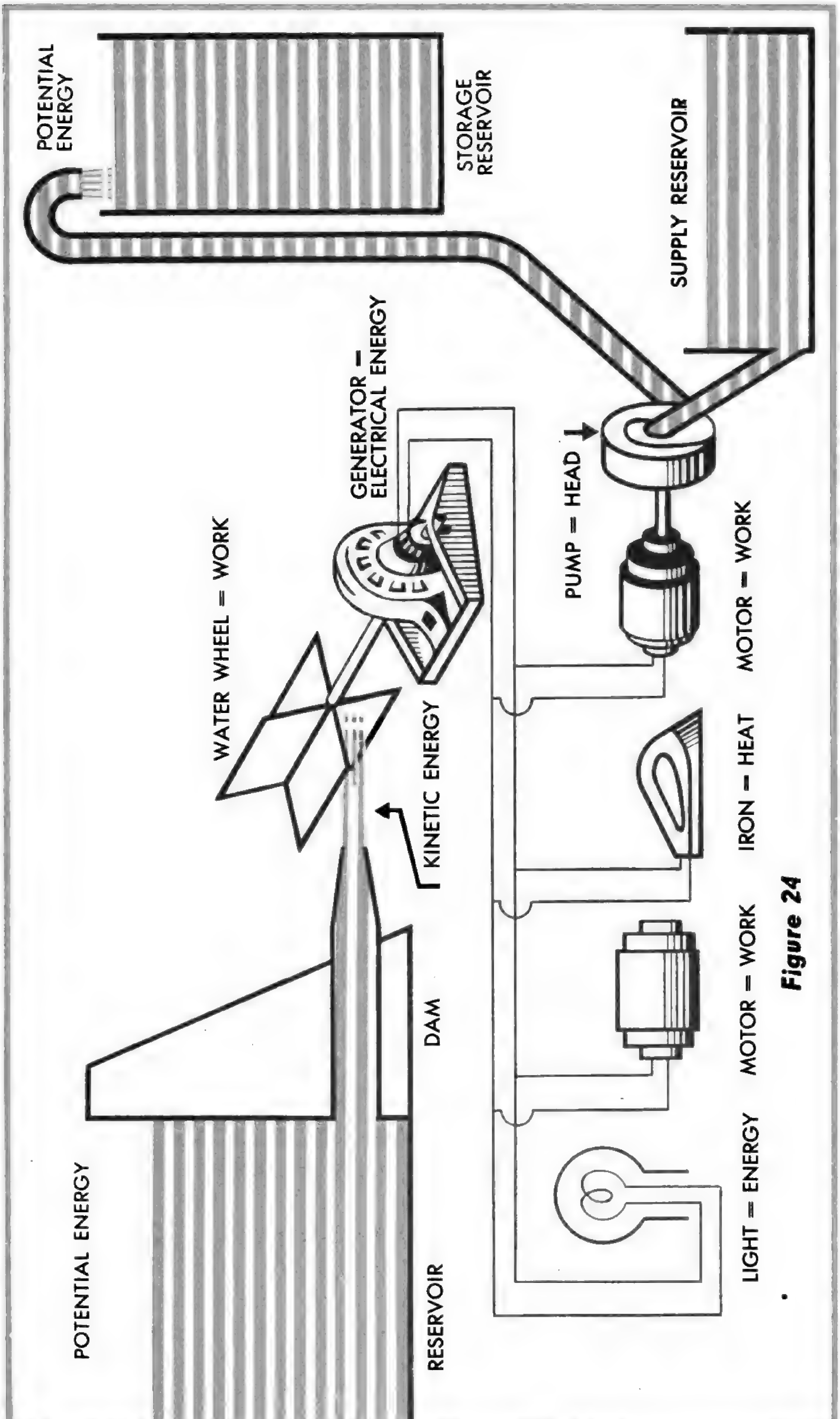


Figure 24

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5. State Pascal's Law.

6. What effect does the shape of the container have on pressure relations in a liquid?

7. What is *head*?

8. What is meant by *specific gravity*?

9. What is the weight of a column of oil 20 inches high whose cross section area is 10 square inches and which weighs 0.03 pounds per cubic inch?

   6 pounds.

10. What is the pressure exerted on the bottom of a container by a column of liquid 100 inches high, whose cross section area is 30 square inches and which weighs 0.03 pound per cubic inch?

    3 pounds per square inch.

11. If the column of liquid in Question *10* stands at sea level, what is the total pressure due to liquid and to atmospheric head on the bottom of the container?

    17.7 pounds per square inch.

12. What is the difference between pressure and force?

13. How can forces be multiplied in a hydraulic system?

14. A hydraulic jack has a single output piston with an area of 100 square inches. The area of the input piston is 10 square inches. If a force of 5 pounds is exerted on the input piston, how much force will be exerted on the output piston?

    50 pounds.

15. If the input piston in Question *14* is moved downward 5 inches, how far up will the output piston be moved?

    0.5 inch.

16. A multiple piston hydraulic jack has two output pistons, each with an area of 50 square inches. The input piston, whose area is 5 square inches, is moved downward 20 inches by a force of 100 pounds. How much pressure is developed in the system? How much force will be exerted on each output piston?

    20 pounds per square inch;
    1000 pounds.

17. What is meant by differential areas on a piston?

18. Why are valves necessary in a hydraulic jack?

19. What is the relation between input and output in a hydraulic system?

20. What is meant by *work*? by *energy*?

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Chapter 2

LIQUID FLOW

This chapter deals with liquids in motion. After a description of what goes on in flow, the forces at work in dynamic situations are considered, and the relations between static and dynamic factors in hydraulic systems are discussed. Various ways of measuring the different factors in flow are described. The chapter concludes with a brief survey of the elements of working hydraulic systems.

Description of Flow

In order to understand hydraulic systems in action, it is necessary to become acquainted with some of the elementary characteristics of liquids in motion. Among these are volume and velocity of flow, steady and unsteady flow, streamline and turbulent flow and, even more important, the force and energy changes that occur in flow and the relations of different kinds of energy to each other in hydraulic systems.

Under normal working conditions the liquid in a hydraulic system is under pressure and completely fills the pipes of the system. This pressure is due in part to the force of gravity or weight of the liquid, and in part to externally applied forces such as might result from the action of a pump, which in turn might have energy supplied to it by an electric motor.

All of the different factors mentioned above will be taken up step by step during the course of this chapter.

Volume and velocity of flow. By volume of flow we mean the quantity of liquid that will pass a given point in a hydraulic system in a unit of time. Volume of flow can be stated in a number of ways, as for example 100 cubic feet per minute, 100 gallons per minute or hour, etc. Gallons per minute is the usual way of expressing volume of flow.

Velocity of flow means the rate or speed at which the liquid is moving forward at a particular point in the system. It too can be variously stated, but the usual method is in feet per second.

Volume and velocity of flow are often considered together. With other conditions unaltered—that is, with volume of input unchanged—the velocity of flow increases as the area of cross section or size of the pipe decreases, and velocity of flow decreases as the cross section area increases. In a stream, velocity of flow will be slow at wide parts of the stream and rapid at narrow parts, even though the volume of water passing each part of the stream is the same. In Figure 25 if the area of cross section of the pipe is 16 square inches at point *A* and 4 square inches at point *B*, the velocity of flow at *B* will be four times the velocity at *A*.

Steady and unsteady flow. A liquid may flow at a single continuous stream, or the volume of flow may increase, decrease or fluctuate

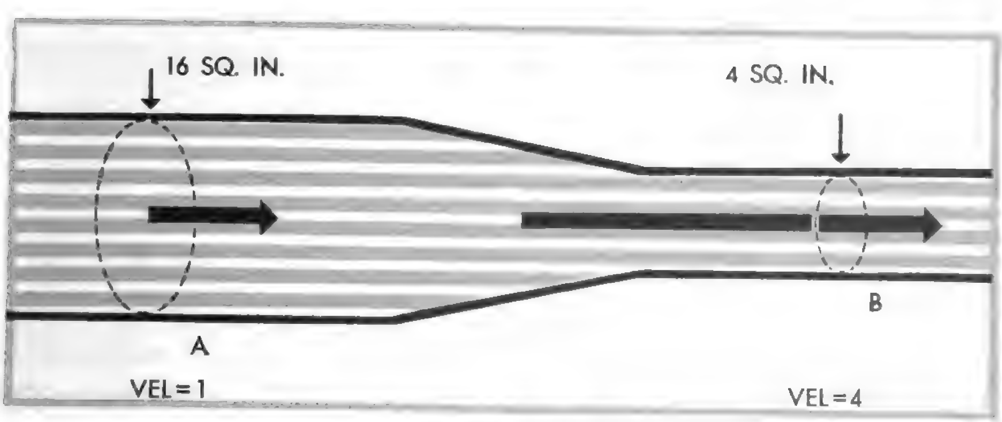

Figure 25

from moment to moment. Such changes in volume constitute unsteady flow. Thus when we turn on a faucet the initial flow will be unsteady during the short time that the rate of flow of the liquid is increasing from the initial zero rate to the full rate of flow. The flow will then become steady and will be maintained if the pressure remains constant. If the pressure changes, the rate of flow once more becomes unsteady until a new balance is reached.

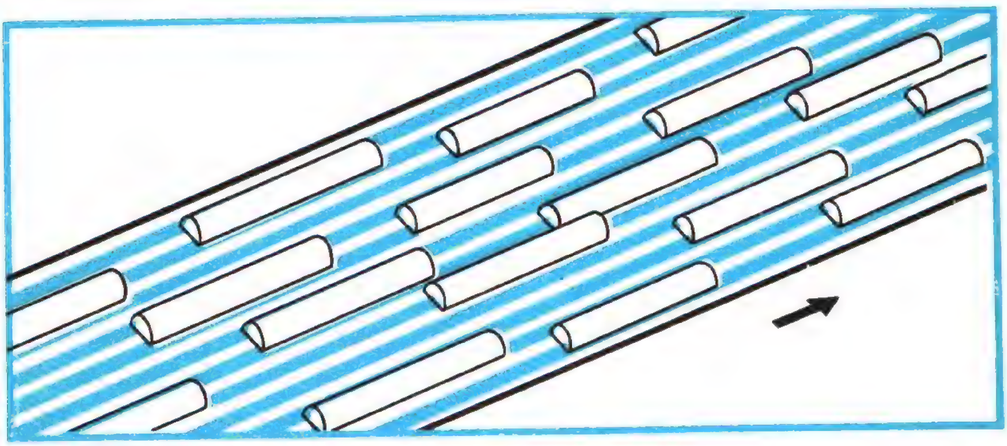

Figure 26

Streamline and turbulent flow. At quite low velocities or in tubes of small diameter, flow will be streamline, meaning that a given particle of liquid will move straight forward without crossing the paths followed by other particles, and without bumping into them. As an example of streamline flow, let us consider Figure 26, where we have an open stream flowing at a slow uniform rate with logs floating on its surface. The logs can be taken to represent particles of water. So long as the stream flows along at a slow uniform rate, each log will float downstream in its own path, without crossing or bumping the other logs.

If the stream narrows, however, and the volume of flow remains the same, the velocity of flow will increase. If the velocity increases sufficiently (Figure 27), the water will become turbulent. Swirls, eddies and cross-motions will be set up in it. As this happens, the logs will be thrown against each other and against the banks of the stream, and the paths followed by different logs will cross and recross.

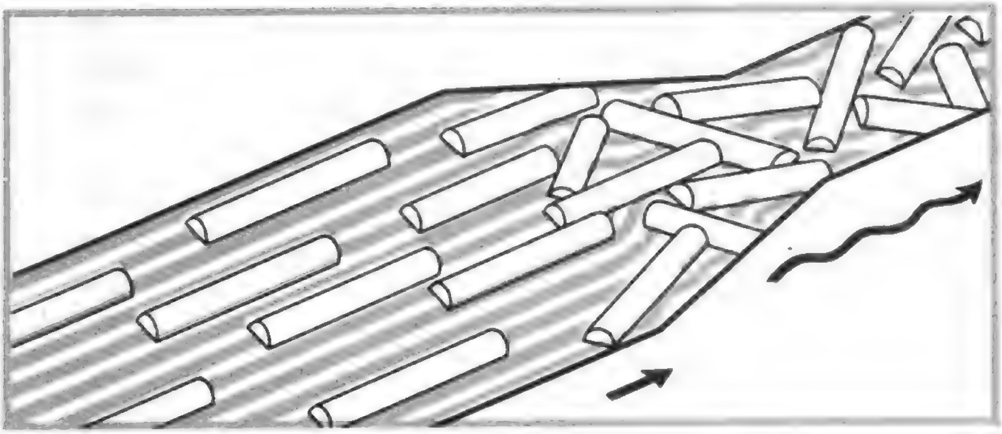

Figure 27

Particles of liquid flowing in pipes act in the same manner. The flow will be streamline if the liquid flows slowly enough, and will remain streamline at greater velocities if the diameter of the pipe is small. If the velocity of flow or size of the pipe increases sufficiently, the flow will become turbulent.

One effect of turbulent flow is shown in Figure 28, where the length of the horizontal arrows indicates the relative velocities of flow at different places in a pipe, from the center to the edge, when the flow is streamline and when it is turbulent. In both instances the rate of flow varies from the center of the pipe to the edge, but streamline flow varies more in velocity than turbulent flow. For streamline flow the average velocity is about half the maximum velocity, while for turbulent flow it is about four-fifths. Velocity of

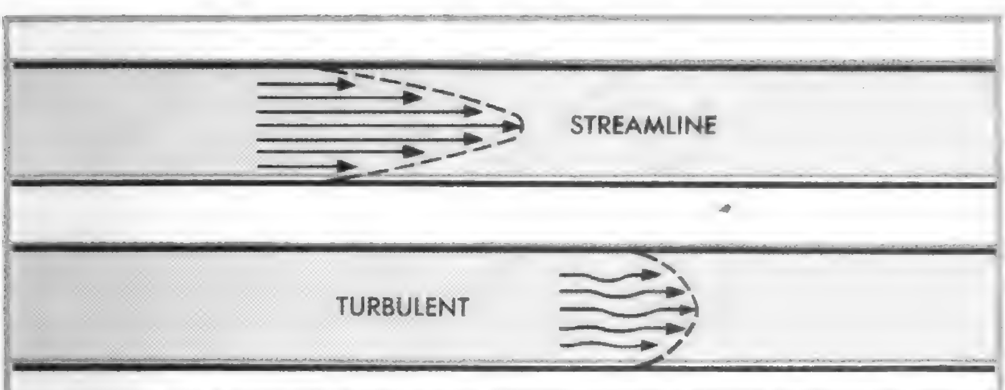

Figure 28

flow of course varies both vertically and horizontally, or from the center of the pipe outwards. In both streamline and turbulent flow, the liquid lying next to the pipe will have little or no velocity.

While a sufficiently great velocity of flow will produce turbulence in any pipe, other factors than velocity contribute to turbulence. Among them are the roughness of the pipe, obstructions, and the degree of curvature of bends and the number of bends in the pipe. In setting up or maintaining a hydraulic system, care should be taken to eliminate or minimize as many causes of turbulence as possible, since the energy consumed by turbulence is wasted. The pipe should be clean and smooth on the inside, and should contain as few bends as possible. In pipe bends the smallest degree of turbulence occurs when pipe is bent on a radius of $2\frac{1}{2}$ to 3 times the inside diameter of the pipe, as shown in Figure 29. Pipe bends must be smooth and uniform, and the inside diameter of the pipe at the bend should be the same size as all other parts of the pipe. A pipe bender should be used if possible, or the pipe should be filled with sand while making the bend.

While designers of hydraulic equipment do what they can to minimize turbulence, to a very considerable extent it cannot be avoided. Thus in a four inch pipe at 68° *F*, flow becomes turbulent at velocities over about six inches per second—or over about three inches

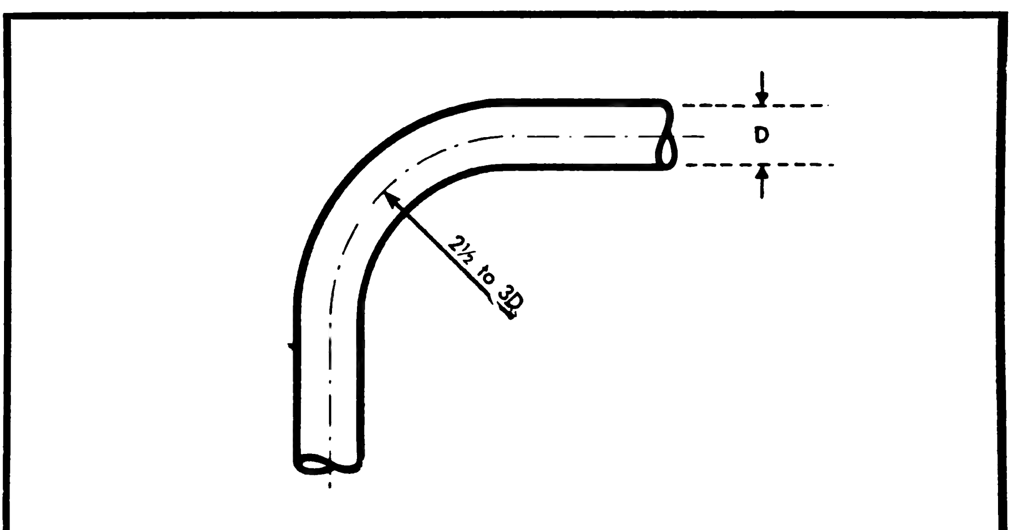

Figure 29

per second for a six inch pipe. These velocities are far below those commonly met in hydraulic systems, where velocities of more than five feet per second are frequent. In streamline flow losses due to friction increase directly with the velocity, while with turbulent flow these losses increase much more rapidly.

Factors Involved in Flow

Inertia. Before we can understand the behavior of liquids in motion, or of solids either for that matter, it is necessary to understand what is meant by inertia. Inertia is the term used by scientists to describe that property possessed by all forms of matter which makes the matter resist being moved if it is at rest, and likewise resist any change in its rate of motion if it is moving.

The basic statement covering the action of inertia is: "A body at rest tends to remain at rest, and a body in motion tends to continue in motion with the same velocity and in the same direction." This is simply saying in more scientific terms what everyone has learned by experience—that one must push on an object to get it moving and offer an opposition to stop it again.

A familiar illustration is the effort a pitcher must exert to make a fast pitch and the opposition the catcher must put up to stop the ball. Similarly, considerable work must be done by the engine of an automobile to get the car started, although after it has attained a certain velocity it will roll along the road at a uniform speed if just enough effort is expended to overcome friction, while brakes are necessary to stop its motion. Inertia also explains the kick or recoil of guns and the tremendous striking force of bullets and shells.

Relation between inertia and force. In order to overcome this tendency to resist any change in its state of rest or motion, some force which is not otherwise cancelled or balanced must act upon the object. Some unbalanced force must be applied whenever liquids are set in motion or speeded up, while conversely, forces are made available to do work elsewhere whenever liquids in motion are retarded or stopped.

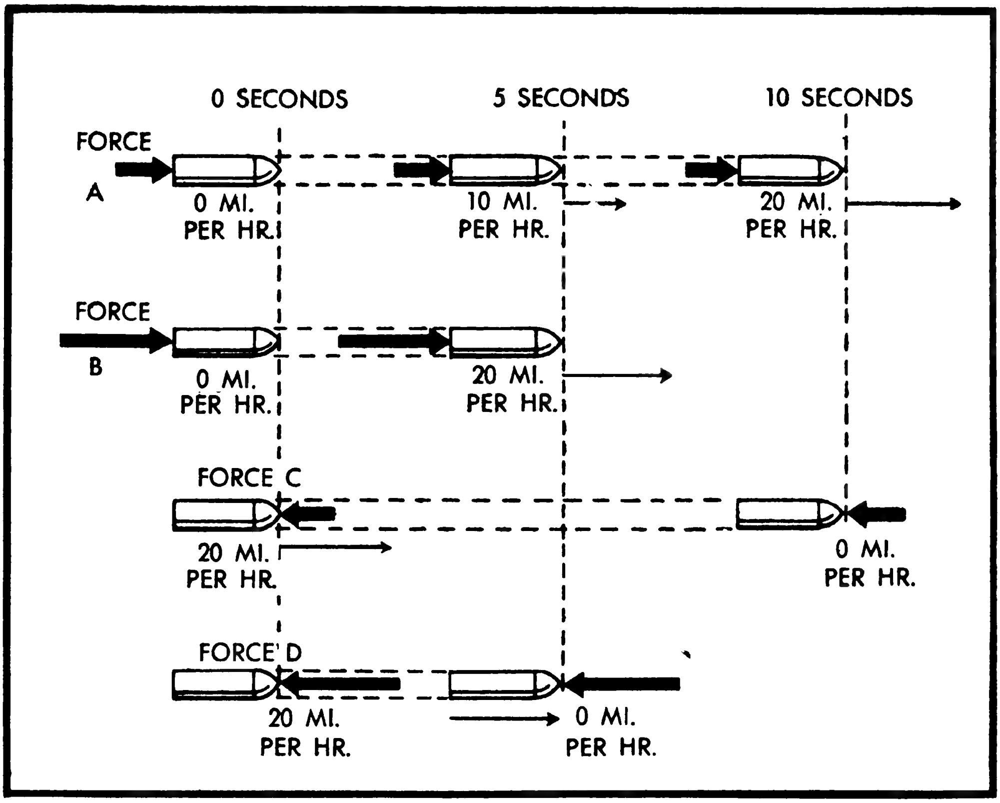

Figure 30

In Figure 30, ignoring friction, if the force *A* will produce a velocity of 10 miles per hour when it is applied to a body for 5 seconds, it will produce a velocity of 20 miles per hour when it is applied for 10 seconds. The same result of 20 miles per hour would be attained if a force *B* equal to twice *A* were applied to the body for 5 seconds. Again ignoring friction, the body would be returned to rest from a velocity of 20 miles per hour if force *C*, equal to *A* but acting in the opposite direction, were applied to it for 10 seconds, or if a force *D* equal to twice *C* were applied to it for 5 seconds.

There is a direct relationship between the magnitude of the force exerted and the inertia against which it acts. This force is dependent on two factors: on the mass of the subject (which is proportional to its weight), and on the rate at which the velocity of the object is changed. While the mathematical relationship between inertia and force is outside the scope of this book, it is included here for completeness and for those who may be interested. The rule

is that the force in pounds required to overcome inertia is equal to the weight of the object, multiplied by the change in velocity measured in feet per second, and divided by 32.2 times the time in seconds required to accomplish the change. Thus the rate of change in velocity of an object is proportioned to the force applied. The number 32.2 appears because it is the conversion factor between weight and mass.

Factors governing hydraulic action. In Chapter 1 we learned that liquids are always acted upon by the force of gravity, or in other words by their own weight. We also learned that they are acted upon by atmospheric pressure, or the weight of the air over the system, if they are exposed to it—if, that is, the system is not completely enclosed. The action of specific applied forces was also explained, and in addition it was brought out that whenever there is movement there is always some friction. We have just introduced a new factor—inertia—which completes the list of forces controlling the action of liquids at rest and in motion.

There are just five physical factors which can act upon a liquid to affect its behavior. All of the physical actions of liquids in all possible systems are determined by the relationships of these five factors to each other. Summarizing, these factors are:

Gravity, which acts at all times upon all bodies, regardless of all other forces;

Atmospheric pressure, which acts whenever any part of a system is exposed to the open air;

Specific applied forces, which may or may not be present, but which in any event are entirely independent of the presence or absence of motion;

Inertia, which comes into play whenever there is a change from rest to motion or the opposite, or whenever there is a change in direction or in rate of motion;

Friction, which is always present whenever there is motion.

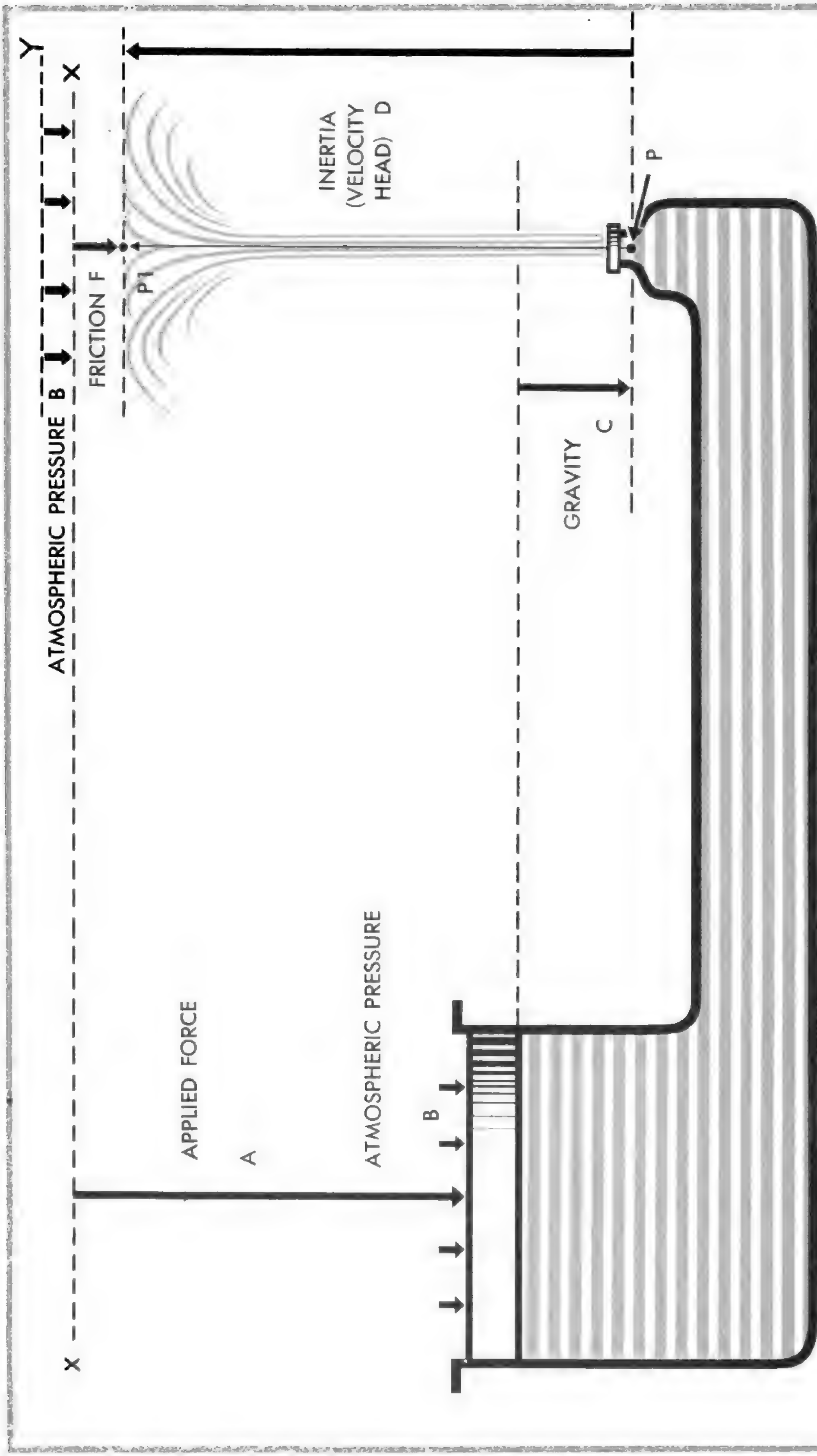
Y
X
ATMOSPHERIC PRESSURE B
FRICTION F
P1
INERTIA
(VELOCITY
HEAD) D
P
GRAVITY C
APPLIED FORCE
A
ATMOSPHERIC PRESSURE
B
X

Figure 31

Figure 31 diagrams a possible relationship of these factors with respect to a particle of liquid *P* in a system. The different forces are shown in terms of head, or in other words in terms of the vertical columns of liquid required to produce the forces. This meaning of *head* was explained in Chapter 1. At the particular moment under consideration, a particle of water *P* is being acted upon by an applied force equivalent to a head of *A*, by atmospheric pressure equivalent to a head of *B*, and by gravity head *C* produced by the weight of the liquid standing over it. The particle possesses sufficient inertia or velocity head to rise to the level *P1*, since head equivalent to *F* was lost in friction as *P* passed through the system. Since atmospheric pressure *B* acts downward on the system on both sides, what was gained on one side was lost on the other.

If all the pressure acting on *P* to force it through the nozzle could be recovered in the form of elevation head, it would rise to the level *Y*; or, if account be taken of the balance in atmospheric pressure, in a frictionless system it would rise to the level *X*, or precisely as high as the sum of gravity head and the head equivalent to the applied force.

Kinetic energy. It has been pointed out above that a force must be applied to an object in order to impart velocity to it or to increase the velocity it already has. Of necessity the force must act while the object is moving over some distance. But we learned in Chapter 1 that a force acting over a distance is *work*, and that work and all forms into which it can be changed are classified as energy. Obviously, then, energy is required to give an object velocity. The greater the energy used, the greater the velocity will be.

Likewise, disregarding friction, for an object to be brought to rest or its motion slowed down, a force opposed to its motion must be applied to it. This force also acts over some distance. In this way energy is given up by the object and delivered in some form to whatever opposes its continued motion. The moving object is therefore a means of receiving energy at one place (where it is speeded up) and delivering it to another point (where it is stopped or re-

tarded). While it is in motion it is said to contain this energy as energy of motion or *kinetic energy.*

We also learned in Chapter 1 that energy can never be destroyed. It follows that if friction be neglected the energy delivered upon stopping the object will exactly equal the energy put in when speeding it up. At all times the amount of kinetic energy possessed by an object depends upon its weight and the velocity at which it is moving.

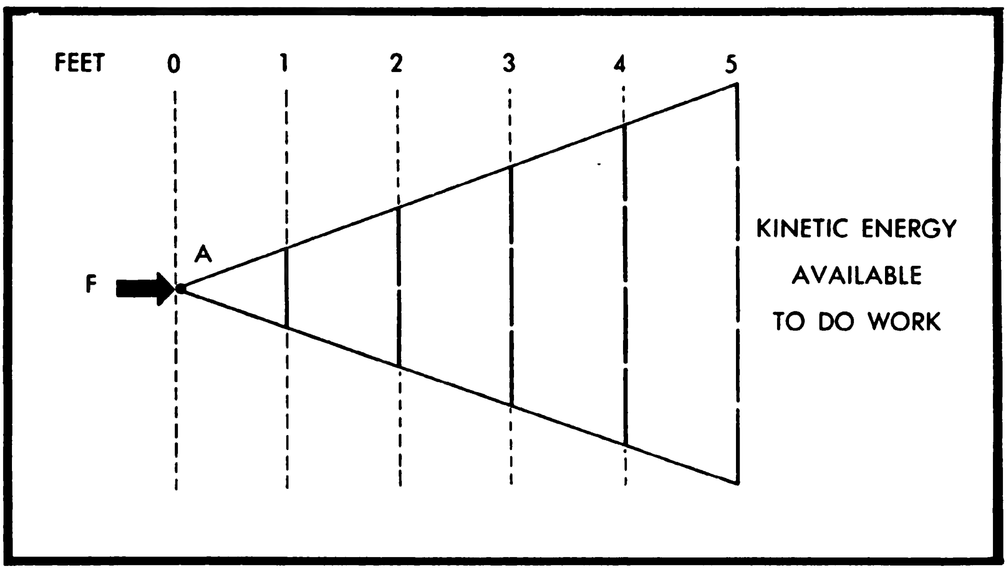

Figure 32

Thus in Figure 32, the force *F* is applied to the body *A*, which is at rest. Disregarding friction, after it has moved one foot it will possess kinetic energy equivalent to *1.* During each succeeding foot of movement it will gain an equal increment of kinetic energy, so long as the force is applied. If it meets a resistance after moving five feet, kinetic energy equivalent to *5* will be available to do work. Accelerated motion has been a means of receiving energy while force *F* was applied to *A*, and of delivering it to do work at the place *A* reached at that time.

For those who may be interested, the mathematical relationship for kinetic energy is stated in the rule, "Kinetic energy in foot pounds is equal to the force in pounds which created it, multiplied by the

distance through which it was applied; or to the weight of the moving object in pounds, multiplied by the square of its velocity in feet per second, and divided by 64.3."

The relationship between inertia forces, velocity and kinetic energy, can be illustrated by analyzing what happens when a gun fires a projectile against the armor of an enemy ship (Figure 33). The explosive force of the powder in the breach pushes the shell out of the gun, giving it a high velocity. But because of its inertia the shell offered opposition to taking this velocity so suddenly, and likewise a reaction was set up which pushed the gun backward (kick or recoil). The force of the explosion acted on the shell throughout its movement in the gun (force acting through a distance producing work). This work appears as kinetic energy in the speeding shell. The resistance of the air produces friction, using up some of the energy and slowing down the projectile. Eventually, however, the shell hits its target and because of its inertia tries to keep on going. The target being relatively stationary tends to stay that way because of its inertia. The result is that a tremendous force is set up which either leads to the penetration of the armor or the shattering of the shell to bits. The shell is simply a means of transferring energy, in this instance for a destructive purpose, from the gun to the enemy ship. This energy is carried in the form of energy of motion or kinetic energy.

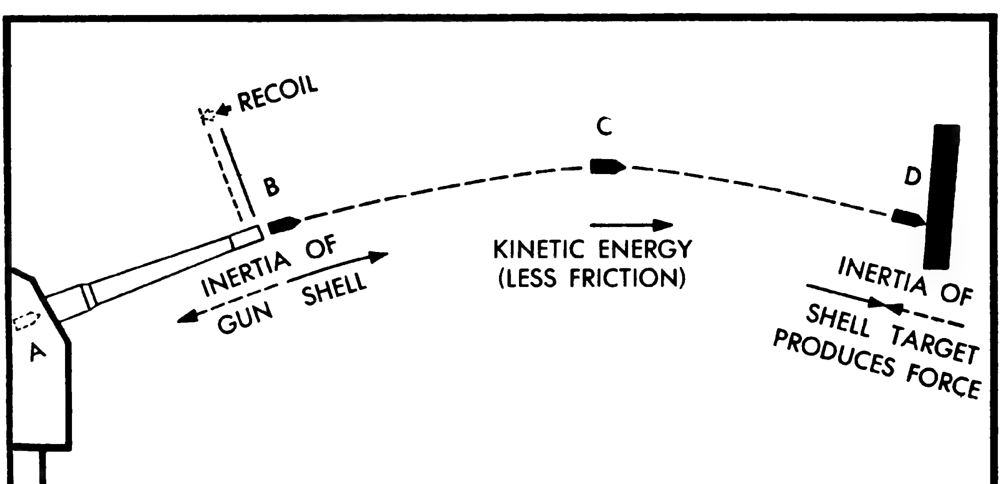

Figure 33

The situation described is shown in Figure 33. The shell is shown in four positions: at *A* where it is at rest in the gun, just before firing; at *B*, a short distance beyond the muzzle of the gun, when its kinetic energy is at a maximum; at *C*, midway in its flight, where friction has used up a portion of its original kinetic energy; and at *D* at the moment of impact, where its kinetic energy is suddenly transformed into work by its inertia and the opposed inertia offered by the target. Energy imparted to the shell at *A* has been transferred in the form of kinetic energy to do work at *D*.

In this diagram no effort was made to exhibit the magnitude of the force of gravity acting on the shell. It enters, of course, into the path the shell takes. This situation differs from that shown in Figure 32 in that the propelling force there was continuously applied throughout the period covered by the diagram, whereas here the force was applied only while the shell was moving from *A* to *B*.

Relationship between force, pressure and head. In dealing with liquids, forces are practically always considered in relation to the areas over which they are applied. But, as we learned in Chapter 1, a force acting over a unit area is a pressure, and pressures can alternatively be stated either in that form (i.e., as pounds per square inch) or in terms of *head,* which is the vertical height of the column of liquid whose weight would produce that pressure.

In most of the applications of hydraulics found in the Navy, applied forces will greatly outweigh all other forces, and in most mechanisms the liquid is entirely confined. Under these circumstances it is customary to think of the forces involved in terms of pressures. Since the term *head* will be met frequently throughout hydraulics, it is necessary to understand what it means and how it is related to pressure and force.

Pressure and head relations in flowing liquids. All five of the factors which control the actions of liquids can, of course, be expressed either as forces, or in terms alternatively of equivalent pressures or heads. In each situation, however, the different factors are commonly referred to in the same terms, since on this common

basis we can add and subtract them and otherwise study their relationship to each other.

At this point some terms in general use should be explained. Gravity head, when it is of sufficient importance to be considered at all, is sometimes known simply as head; the effect of atmospheric pressure is frequently and improperly referred to as suction (see Chapter 1 for explanation); inertia effect, because it is always directly related to velocity, is usually called velocity head; and friction, because it represents a loss of pressure or head, is usually referred to as friction head.

Static and dynamic factors. The first three factors—gravity, applied forces, and atmospheric pressure—apply equally to liquids at rest or in motion, while the latter two—inertia and friction—apply only to liquids in motion. The first three are the static factors and the latter two are the dynamic factors. The arithmetic sum of the first three—gravity, applied force, and atmospheric pressure—is the static pressure obtained at any one point in a liquid at a given time. Static pressure exists in addition to any dynamic factors which may also be present at the same point and time.

In Chapter 1 it was learned from Pascal's Law that a pressure set up in a liquid acts equally in all directions and at right angles to containing surfaces. This covers the situation only for liquids at rest, or practically at rest. It is true only for the factors making up static head. It was for that reason that we stated that friction was disregarded in all the problems of Chapter 1. Obviously, when velocity becomes a factor it must have a direction, and as already explained the force related to the velocity must also have a direction, so that Pascal's Law alone does not apply to the dynamic factors of liquid flow.

Relation of static and dynamic factors. The dynamic factors of inertia and friction are related to the static factors in one sense, however. Velocity head and friction head are obtained at the expense of static head. On the other hand at least a portion of velocity head can always be reconverted to static head. As we

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already know, force, which can be produced by pressure or head when we are dealing with liquids, is necessary to start a body moving if it is at rest, and is present in some form when the motion of the body is arrested. In other words, whenever a liquid is given a velocity, some part of its original static head is used to impart this velocity, which then exists as velocity head.

Let us now consider a system consisting of a chamber *A* under pressure connected by a tube to chamber *B*, which is also under pressure (Figure 34). The pressure in chamber *A* will be wholly static pressure, say 100 pounds per square inch. The pressure at some point *X* along the connecting tube will consist of a velocity pressure of say 10 pounds per square inch exerted in a direction parallel to the line of flow, plus the unused static pressure of 90 pounds per square inch, which still obeys Pascal's Law and operates equally in all directions. As the liquid enters chamber *B* it is slowed down, and in so doing its velocity head is changed back into pressure head. In other words, the force required to absorb its inertia equals the force required to get the liquid moving in the first place, so that the static pressure in chamber *B* will again be equal to that in chamber *A*, although it was lower at an intermediate point.

The situation described disregards friction, and would therefore not be met in actual practice. Friction also requires force or head to overcome it, but, contrary to the inertia effect, this force cannot

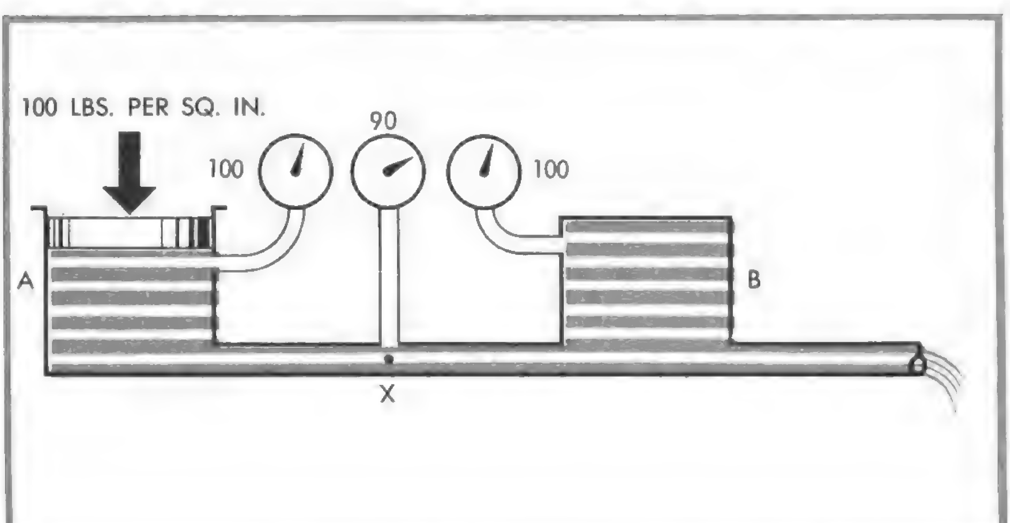

Figure 34

be recovered again, although the energy represented still exists somewhere as heat, as explained in Chapter 1. Therefore, in an actual system the pressure in chamber *B* would be less than that in chamber *A* by the amount of pressure used in overcoming friction along the way.

At all points in a system, therefore, the static pressure will always be the original static pressure less any velocity head at the point in question, and less the friction head consumed in reaching that point. Since both velocity head and friction head represent energy which came from the original static head, and since energy cannot be destroyed, the sum of the static head, velocity head and friction head at any point in a system must add up to the original static head. This general truth is known as Bernoulli's Theorem, and is the second important basic law of hydraulics. It governs the relations of the static and dynamic factors, while Pascal's Law states the manner in which the static factors behave when taken by themselves. Figure 31 illustrated this point.

Minimizing friction. Hydraulic equipment is designed to keep friction at the lowest possible level. Volume and velocity of flow are made the subject of careful study. The proper liquid for the system is chosen. Clean smooth pipe of the best dimensions for the particular conditions is used, and it is laid along as direct a route as possible. Sharp bends and sudden changes in section area are avoided. Valves, gauges and other obstructions are designed so as to interrupt the flow as little as possible. Careful thought is given to the size and shape of openings. The system is designed so that it can be kept clean inside and so that variations from normal operation can easily be detected and remedied.

Measurement of Flow Factors

We are now in a position to discuss measurement of the factors involved in flow. For purposes of comparison they should all be stated in the same units, as for example feet of head, inches of head, pounds per square inch, etc. We shall write this section in terms of head rather than of pressure, since head lends itself more easily to the making of illustrative diagrams when gravity is under

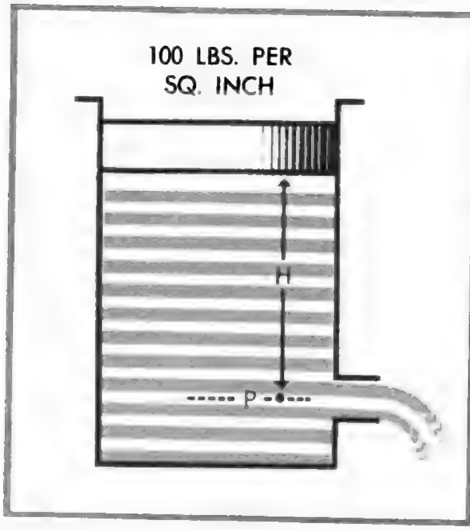

Figure 35

consideration. In our examples atmospheric pressure will be ignored, since it would balance out in the situations we shall consider.

Measurement of input head. Input head due to gravity is measured by the vertical distance from the surface of the liquid to some convenient horizontal reference line. When the liquid is also subject to an applied force, the head equivalent to this force must be added. Thus in Figure 35, where the water in the tank is under a force yielding a pressure of 100 pounds per square inch, the total input head at P is $H +$ 230.9 feet, since it would take a column of water 230.9 feet high to produce a pressure of 100 pounds per square inch.

Input head measures the total potential energy available for hydraulic use. It can be measured between any two points in a system, as was done in Figure 35, or between any point in a system and a convenient reference level. In each case it measures the energy available to do work between the levels used.

Measurement of static pressure head. In Figure 36, liquid is standing so that it just fills a pipe. The pressure at A is obviously zero. If the level of standing liquid were as shown in Figure 37, pressure at A would be as shown. With standing liquids, the same pressure level will be shown, no matter what the size, shape or inclination of the tube. Thus the tube at B shows the same pressure as the tube at A. With moving liquids, however, static pres-

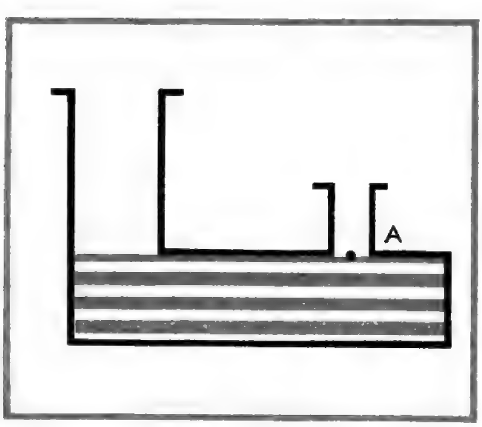

Figure 36

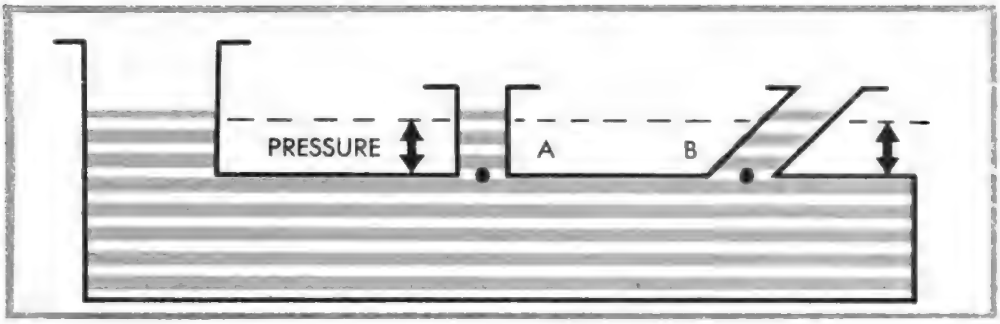

Figure 37

sure can be measured only by a tube standing at right angles to the flow in the system.

If a measuring tube is inclined in the direction of flow, the level in the tube will stand too high; if it is inclined against the direction of flow, it will stand too low (Figure 38). This is because the velocity head, which acts in one direction only, tends to add to the pressure head at *C* and to subtract from it at *A*. The tubes must be flush with the inner surface of the pipe, and smoothly joined to it, so as to minimize friction.

Measurement of velocity head. If we know the rate of volume flow of a liquid, and the area of cross section of the pipe at a given point, the velocity of flow there is equal to the first figure divided by the second. Velocity can then be stated as velocity head by remembering that head varies as the square of velocity. For the sake of those who may be interested, the numerical values can be determined by the

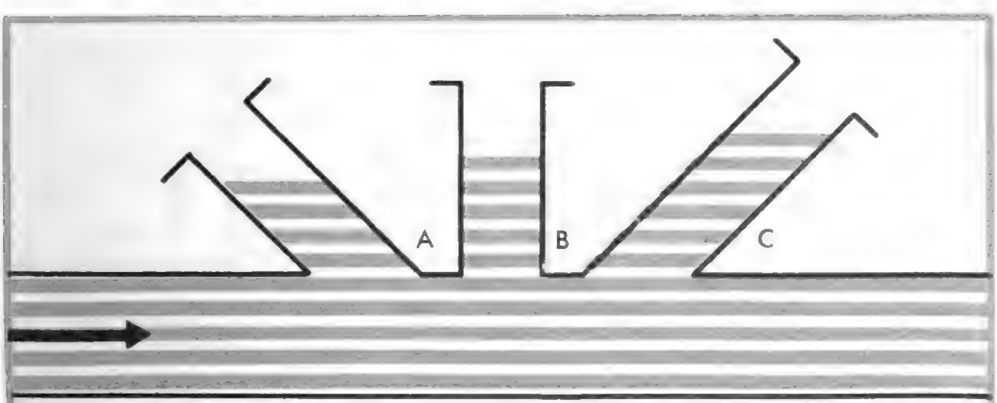

Figure 38

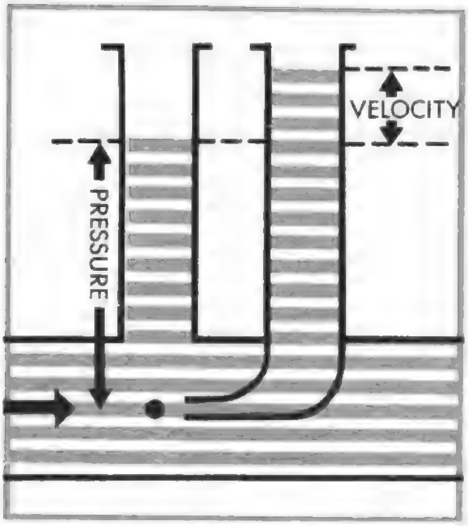

Figure 39

following formula:

$$\text{Head in feet} = \frac{(\text{velocity in feet per second})^2}{64.3}$$

Thus if water is flowing at the rate of 300 cubic feet per minute, and the area of cross section of the pipe at a certain point is 0.4 square foot, the velocity there will be 750 feet per minute (300/0.4), or 12.5 feet per second (750/60). Using the above formula, the velocity head in feet will be $12.5^2/64.3$ or 2.44 feet.

Velocity head can be directly measured at any point in a pipe by inserting a Pitot tube into the pipe at the point desired (Figure 39). This is a small tube open at both ends and curved at the bottom to minimize friction when the tube is inserted into a moving liquid. The opening of the tube is inserted directly in the line of flow so that the liquid running into it has its velocity head converted into the equivalent static head. The tube also measures the static pressure prevailing in the liquid, which must be subtracted to give velocity head. Static pressure can be obtained by another tube inserted at right angles to the direction of flow.

If the Pitot tube is turned so that the opening is at right angles to the

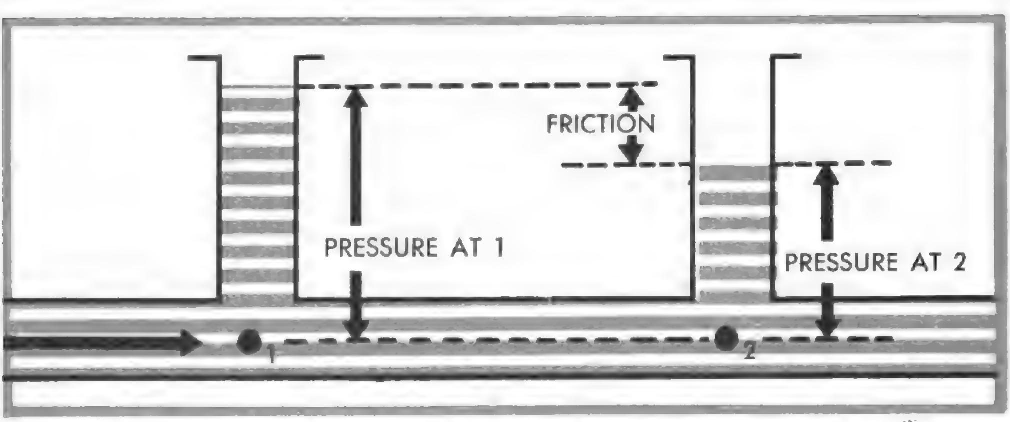

Figure 40

flow, the liquid in the tube will drop to the static pressure head level. If the opening is placed directly opposite to the line of flow, the liquid in the tube will stand below the static pressure level by an amount approximately equal to the velocity head.

Measurement of friction head. In Figure 40, the static pressure heads at *1* and at *2* are as shown. The friction head between *1* and *2* is therefore the difference, assuming that the velocity at both points is the same.

Although the friction head as between *1* and *2* was here measured in terms of static pressure, the magnitude of the total static head apart from the friction will have no effect whatever on the friction head. Thus in Figure 41, where the liquid in the pipe is now under a considerably greater static pressure, the friction head between the two points remains unchanged for the same pipe, liquid and velocity.

Water Hammer

When flow in a pipe is too quickly interrupted, as for example by the hasty closing of a valve, the onward motion of the liquid is suddenly halted. The inertia of the liquid causes a sudden high pressure, the effect of which is like a hammer blow. This sets up a pressure wave moving at about a mile per second surging back and forth

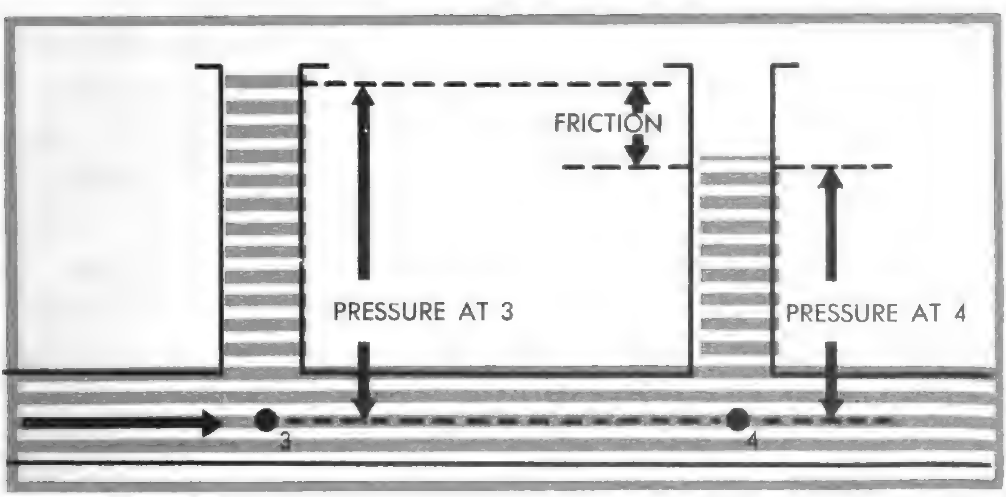

Figure 41

through the system, alternately stretching and compressing the pipe. This can put a system under great strain; the condition is accompanied by clanking sounds.

The pressure generated by water hammer is over and above the ordinary pressure of the system, since it comes from the practically complete and very sudden stoppage of the liquid in the same manner as the pressure developed by the impact of a shell against armor as previously considered. Care should be taken not to induce water hammer, since it may rupture the system or damage connections. Conditions leading to chronic water hammer should be investigated and remedied, as for example by the installation of slower-acting valves or pipe of larger diameter near to the valves. The addition of an air chamber just ahead of a valve subject to this condition will help to absorb the pressure waves accompanying closure of the valve.

Elements of Working Hydraulic Systems

Indispensable parts of a working system. So far we have discussed hydraulic systems only in rather general terms. We have not, for example, considered the special qualities of the different liquids that might be used, nor the different kinds of pipes and fittings that might be employed to carry the liquid, nor have we accounted for the forces applied. Under actual working conditions a pump is also needed to overcome the inertia of the liquid and supply the energy needed to drive the working parts of the system. A source of energy, as for example steam power or an electric motor, is also needed to run the pump. The energy supplied to the liquid through the pump must be transmitted through pipes to a work cylinder and piston, or the equivalent, and the piston must be mechanically linked to the actual work operation, where some resistance will be overcome. In addition, a reservoir is needed from which liquid can be pumped when necessary, and to which liquid can be returned when the system must be relieved. Finally, valves are required to direct and control the flow of liquid, and gauges to measure the pressure at different points.

Schematic diagrams. Many arrangements of the different parts of a

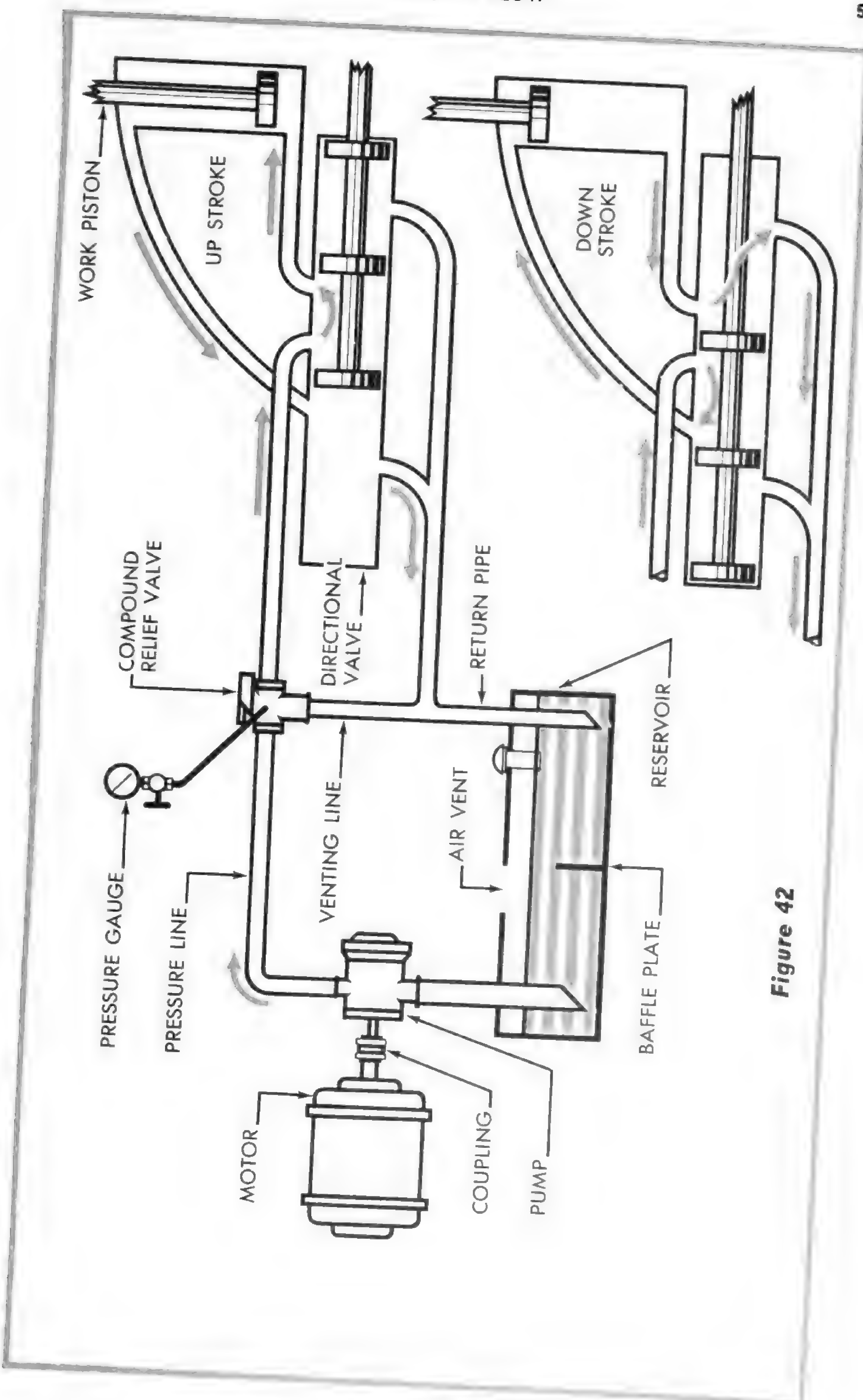
WORK PISTON
UP STROKE
DOWN STROKE
COMPOUND RELIEF VALVE
DIRECTIONAL VALVE
RETURN PIPE
RESERVOIR
PRESSURE GAUGE
PRESSURE LINE
VENTING LINE
AIR VENT
BAFFLE PLATE
MOTOR
COUPLING
PUMP

Figure 42

hydraulic system are possible, and an actual illustration of even a simple system might be quite complicated. The working relations of the parts of a system to each other can be indicated, however, by a *schematic diagram,* without any effort to indicate in detail how each part works, exactly where it is located, or what its actual size is. From study of a schematic diagram, it is possible to see how the system as a whole is organized. Figure 42 gives an example of a schematic diagram.

Complicated hydraulic systems obey all the laws of hydraulics just as rigorously as do the simplest devices. In the chapters that follow, the student should take the trouble to test each new situation as it arises in the light of fundamental hydraulic principles.

QUESTIONS

1. What is the quantity of flow in gallons per minute of a liquid passing through a pipe with a cross section of 10 square inches at a velocity of 0.77 inches per second? (One gallon contains 231 cubic inches.)
   2 gallons per minute.

2. What is the velocity of flow of a liquid passing through a pipe with a cross section of 5 square inches at the rate of 20 gallons per minute?
   15.4 inches per second.

3. What is meant by unsteady flow? By streamline flow? Bv turbulent flow?

4. Why is pressure necessary to cause flow?

5. How much pressure is required to cause flow through a pipe 100 feet long if friction causes a pressure of 2 pounds per square inch for every 5 feet of pipe?
   Over 40 pounds per square inch.

6. What are the five physical factors capable of acting on a liquid in a hydraulic system?

7. What is the relation between inertia, velocity and pressure?

8. What is the relation between input and output factors in a hydraulic system?

9. What factors enter into losses through friction in a hydraulic system?

10. What is the static pressure just beyond the point of exit from a hydraulic system?

11. Can the static pressure in a hydraulic system increase, without an increase in applied force, as one moves along the direction of flow?

12. What is meant by water hammer? How can it be avoided?

13. What are the indispensable parts of a working hydraulic system?

14. What is a schematic diagram?

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Chapter 3

PRESSURE GAUGES AND VOLUME METERS

In this chapter we describe the construction, operation and care of the Bourdon and the Schrader pressure gauges and the Niagara volume meter.

Pressure gauges are used in hydraulics to measure system pressures so that the operators of hydraulic equipment can maintain pressures at safe and efficient operating levels. Any excess or deficiency of pressure should immediately be investigated with a view to locating and removing the cause of the trouble.

Gauge pressure and absolute pressure. When the kind of gauge in ordinary use reads say 30 pounds per square inch, the reading applies to the liquid pressure set up by the opposition of forces within the system. Atmospheric pressure also acts on the system, but it can be ignored in practical operation because its action at one place is balanced by its equal and opposite action at another place. When it is taken into account in scientific calculations, the pressure of the system is referred to as absolute pressure. In this book, however, we confine our attention to gauge pressures.

Bourdon Pressure Gauge

Principle of the gauge. The Bourdon gauge works on the principle that pressure in a curved tube will tend to straighten it out. Thus in Figure 43 pressure acts equally on every square inch of area in the

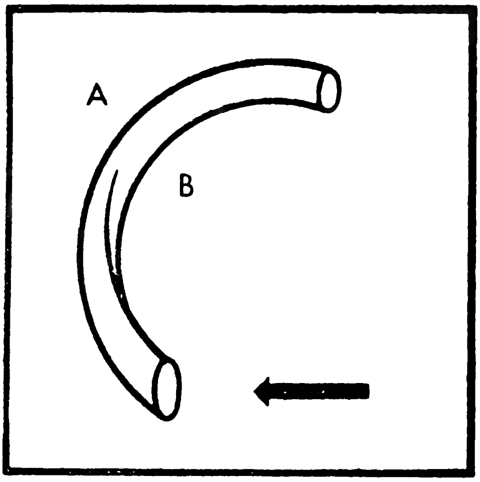

Figure 43

tube, but since the surface area *A* on the outside of the curve is greater than the surface area *B*, on the shorter radius, the force acting on *A* will be greater than the force acting on *B*. When pressure is applied the tube will straighten out until the difference in force is balanced by the elastic resistance of the material composing the tube. If a pointer be attached to the tube and a scale laid out on which the pointer can register pressures, we have the main elements of a Bourdon gauge.

Description of the gauge. Figure 44 shows the working parts of a Bourdon gauge. The tube is bent into a circular arc, and is oval in cross section so that it will tend to straighten more easily when under pressure. The tube works by differential areas, since the area on which the pressure acts outward is greater than the area on which the pressure acts inward. The open end of the tube passes through a socket which is threaded so that the gauge can be screwed into an opening in the hydraulic system. The closed end of the tube is linked to a pivoted segment gear in mesh with a small rotating gear to which a pointer is attached. Beneath the pointer is a scale reading in pounds per square inch. The gauge is calibrated against known pressures to insure accurate readings. The working parts are mounted in a metal case with the dial visible through a glass face (Figure 45.) Under pressure the tube tends to straighten and the segment gear moves about its pivot, rotating the meshing gear and the pointer.

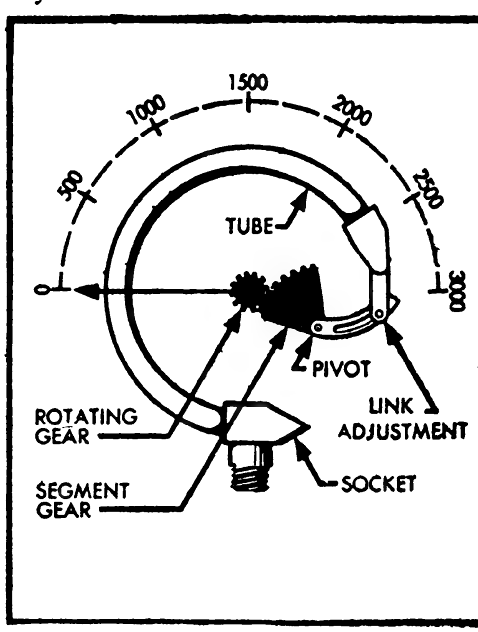

Figure 44

The link adjustment between the segment gear pivot and the end of the tube can be set to make the gauge read correctly for known pressures. The pointer can also be loosened on its shaft and set so that the gauge will stand at zero when no pressure is being applied.

Care and use. The Bourdon gauge is made in various sizes. It can be manufactured cheaply for general use, or it can be more carefully made for precision work. Precision types are satisfactory for measuring high pressures, since they can be made accurate to about 0.25 per cent of the full scale reading. Bourdon gauges of all kinds need frequent resetting. Calibration is accomplished by applying known pressures on a dead weight gauge tester and adjusting the pointer to read accurately.

Figure 45

The gauge should be protected from vibration, from excessive temperatures, from corrosive liquids, and from sudden high pressures. Liquid should be introduced into the gauge by slowly opening the petcock in the gauge line.

Schrader Hydraulic Gauge

The Bourdon gauge is a delicate instrument, and can be damaged very easily. In addition, as the range of hydraulic applications widened to include systems where the pressure fluctuates rapidly, the gauge sometimes caused trouble. The Schrader direct action gauge was therefore developed, and since its introduction, not many years ago, has been increasingly accepted.

Principle of the Schrader gauge. In the Schrader gauge a piston is directly actuated by the liquid pressure to be measured, and moves up a cylinder against the resistance of a spring, carrying a bar or

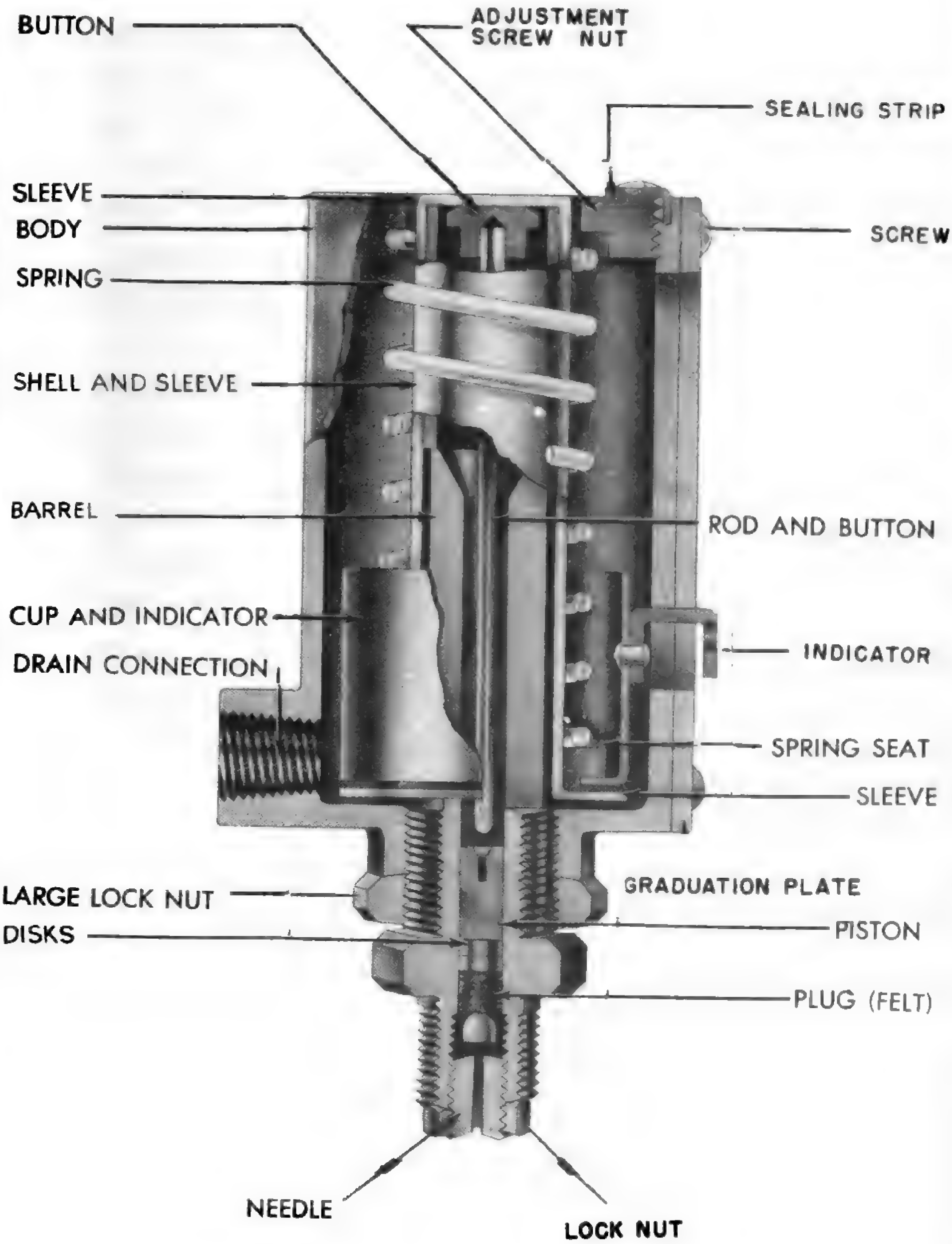
BUTTON
ADJUSTMENT SCREW NUT
SEALING STRIP
SLEEVE
BODY
SCREW
SPRING
SHELL AND SLEEVE
BARREL
ROD AND BUTTON
CUP AND INDICATOR
INDICATOR
DRAIN CONNECTION
SPRING SEAT
SLEEVE
LARGE LOCK NUT
GRADUATION PLATE
DISKS
PISTON
PLUG (FELT)
NEEDLE
LOCK NUT

Figure 46

indicator with it over a calibrated scale. In this manner all levers, gears, cams and bearings are eliminated, and a sturdy instrument can be constructed.

How the gauge operates. Figure 46 shows the construction of the gauge and the manner in which it operates. The parts up through the middle of the gauge, from the needle at the bottom on through the packing piston and rod to the button at the top, form a unit which transmits hydraulic pressure to the sleeve against which the button rests. This sleeve surrounds the inner barrel of the gauge, and is flanged at its base to provide a seat for a spring coiled around the barrel, and for a cup to which the pressure indicator is attached. Hydraulic pressure compresses the spring, and the barrel rises out of the body, carrying the indicator up with it. The indicator moves against a pressure scale on the face of the gauge (Figure 47).

Calibration. The Schrader gauge is calibrated by comparing gauge readings with known pressures. A small error can be corrected by loosening the four screws on the face of the gauge and sliding the scale up or down under the pointer. For larger errors, the adjustment screw which holds the spring in place can be tightened if the

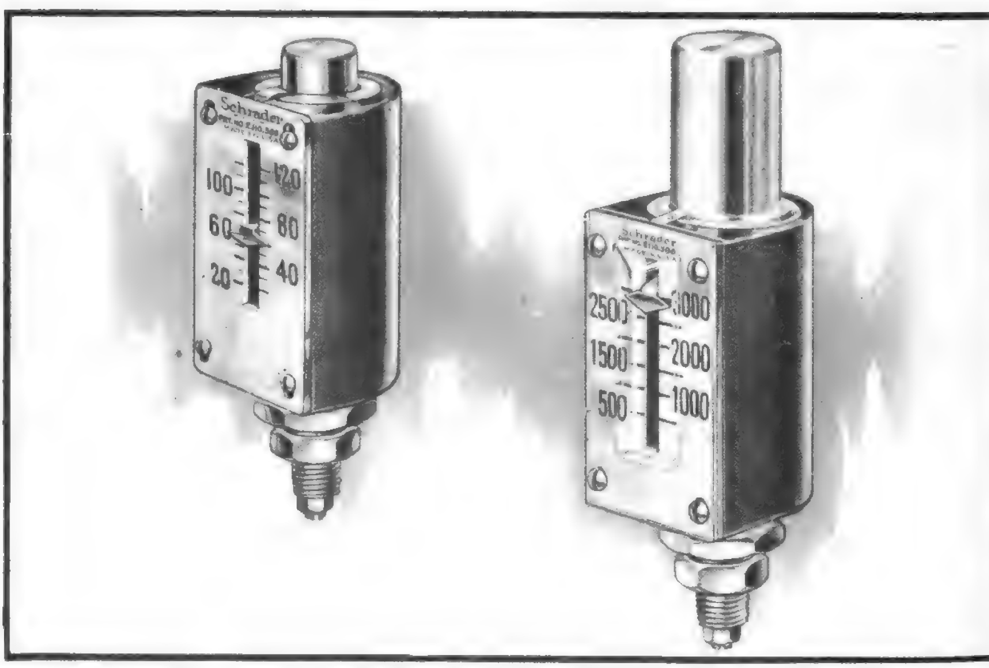

Figure 47

gauge reading is too large, or loosened if the reading is too small. Turning the adjustment screw varies the compression of the spring. A sealing strip is provided to lock the adjustment screw in place after calibration.

Disassembly. The inner core of the gauge can be removed for cleaning or for replacement of parts by loosening the small lock nut at the base and the large lock nut just above it, and then unscrewing the barrel from the body. In removing parts, be sure to observe their relative positions as given in Figure 46. In reassembly, the barrel is rescrewed into the body until the indicator marker moves 1/16 inch, after which the large lock nut is tightened. If the needle at the base is screwed into place too tightly, readings will be sluggish; if too loosely, readings will be jumpy.

The other interior parts can be reached by removing the face plate and then unscrewing the adjustment screw nut, after breaking the solder seal of the sealing strip. The exact position of the adjustment screw nut should be marked on the casting with a knife or chisel, so that it can be properly positioned when the sealing strip is resoldered.

Applications. While the Schrader gauge is satisfactory for ordinary hydraulic pressure measurements, it is not as accurate as the Bourdon gauge. It is not a laboratory gauge for exact readings, but a sturdily constructed working unit suitable for practical use. It is especially to be recommended for fluctuating loads.

The gauge should be protected against vibration, excessive temperatures, corrosive liquids, and sudden high pressures. A petcock should be installed between the gauge and the pressure system so that pressure can be applied slowly to the gauge, and so that it will be protected against strain when not in actual use.

General precautions. Gauges should not be used on a system whose maximum pressure may exceed the maximum designated gauge reading. Dropping a gauge may permanently damage the calibrated units. When gauges are not in use they should be stowed in a dry place.

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Volume Meters

Wherever liquids are consumed some method of checking intake against outgo must be provided. This is particularly true as regards diesel engine and boiler fuels, where the hourly quantitative record of fuel expenditure provides a check against the sounding of fuel-oil tanks and promotes engineering analysis and economy. Meters suitable for naval use must be rugged, durable, accurate, and suitable for prolonged use with little adjustment.

Any pump which displaces a uniform volume of liquid for each stroke of its piston could be used as a meter by installing a device for counting the piston strokes. But the pump would have to be designed to minimize leakage and guarantee uniform displacement.

Niagara meter. The Buffalo Meter Company manufactures a meter with a disk piston which acts on the principle most widely used for measuring liquids in moderate quantities. Meters of this kind are available to measure flow up to 1000 gallons per minute, but their use beyond 400 gallons per minute is unusual. This range covers the great majority of naval applications where an accurate meter is required.

Principle of the disk piston. In this type of meter the liquid passes through a fixed volume measuring chamber divided into upper and lower compartments by the disk. One or the other compartment is continually being filled while the other is being emptied. As it passes through these compartments the pressure of the liquid causes the disk to roll around in the chamber, in a manner to be described, and its movements operate a counter, through suitable gearing, to indicate the volume of liquid passed. The counter somewhat resembles the mileage indicator on an automobile, except that it usually reads in gallons.

The heart of the meter is the measuring chamber and disk piston. Figure 48 shows how the disk piston is located in the measuring chamber. Half of the measuring chamber itself is shown in Figure 49. It is bounded at top and bottom by conical surfaces *1* and *2*,

which are joined at their outer edges by spherical surface *3*. A view of the meter as a whole is given in Figure 50. Referring to Figure 48, the upper and lower surfaces of the chamber converge towards the center to form a spherical cavity for the ball *4*, through which passes the spindle *5*, which is connected with the counting gears (Figure 50). Attached to the ball is the disk *6*. Both ball and disk are machined to fit closely in the chamber. There is a slot *8*

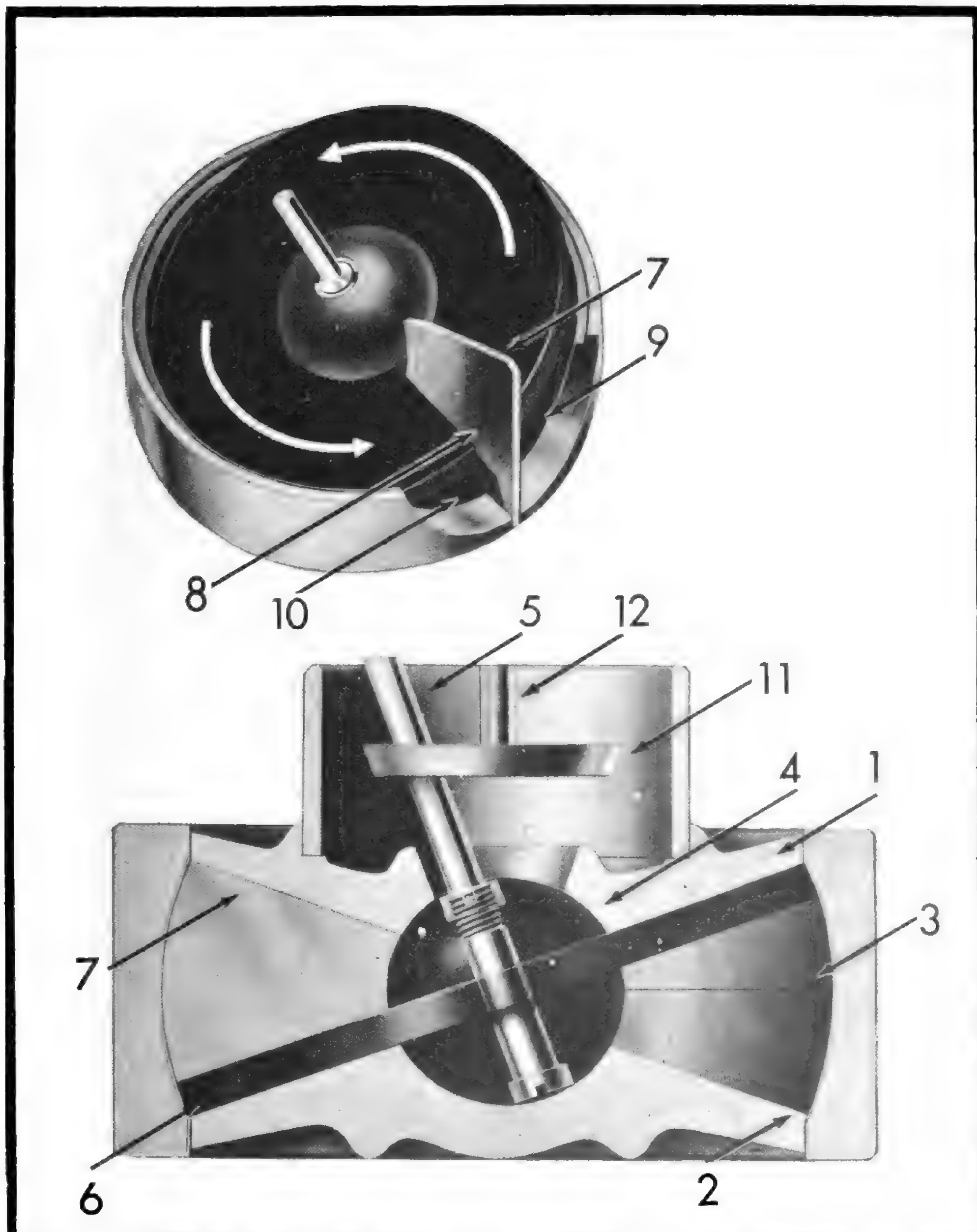

Figure 48

in the disk at one point, through which passes the thin partition *7*, which cuts the chamber in two. There are openings in the outer wall of the chamber on each side of the partition. Opening *9* is the liquid inlet, while *10* is the outlet.

When the ball carrying the disk and the spindle is tilted as far as it can go, the bottom of the disk makes a close contact with the bottom

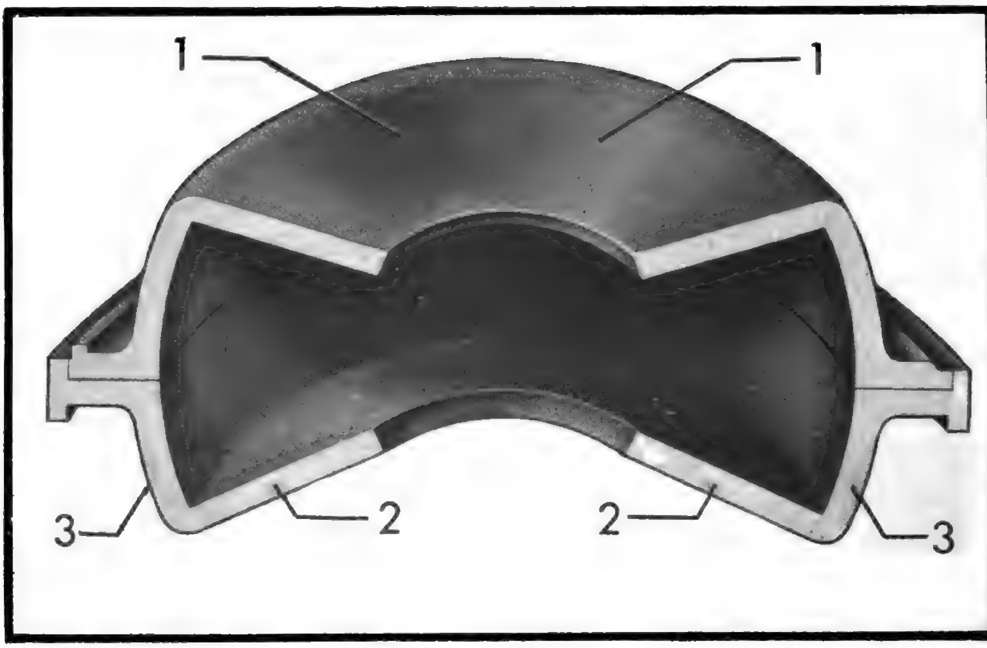

Figure 49

conical surface of the chamber, while the top of the disk similarly contacts the upper conical surface at the opposite end of the disk. Since the disk is flat while the contacting surfaces are conical, contact takes place along a straight line on each surface. The lines of contact produce seals, which when taken in connection with the partition 7, divide both the upper and lower compartments into two parts. The net effect is that the disk and partition produce four separate compartments in the measuring chamber, two above the disk and two below. The top and bottom compartments are separated from each other by the disk itself, while the pairs of compartments respectively above and below the disk are separated by the seal formed at the line of contact between the disk and the conical top and bottom surfaces on the one hand, and by the partition on the other.

Spindle *5* rises from ball *4* and passes through wheel *11* at a point near its outside edge. Vertical shaft *12* is attached to wheel *11* and rotates with it. This shaft is connected at its upper end to the measuring gears, and is mounted directly over ball *4*. When wheel *11* turns on its axis, the position of the shaft keeps the spindle inclined at just the angle to produce a continuous seal between disk *6* and the upper and lower surfaces of the measuring chamber.

The action of the meter can be understood by imagining wheel *11* to be revolved by means of shaft *12*. The spindle would revolve with it, and would trace a conical path in space, as shown in Figure 51. This movement of the spindle would control the position of the disk. When the spindle was in position *A*, for example, the disk would

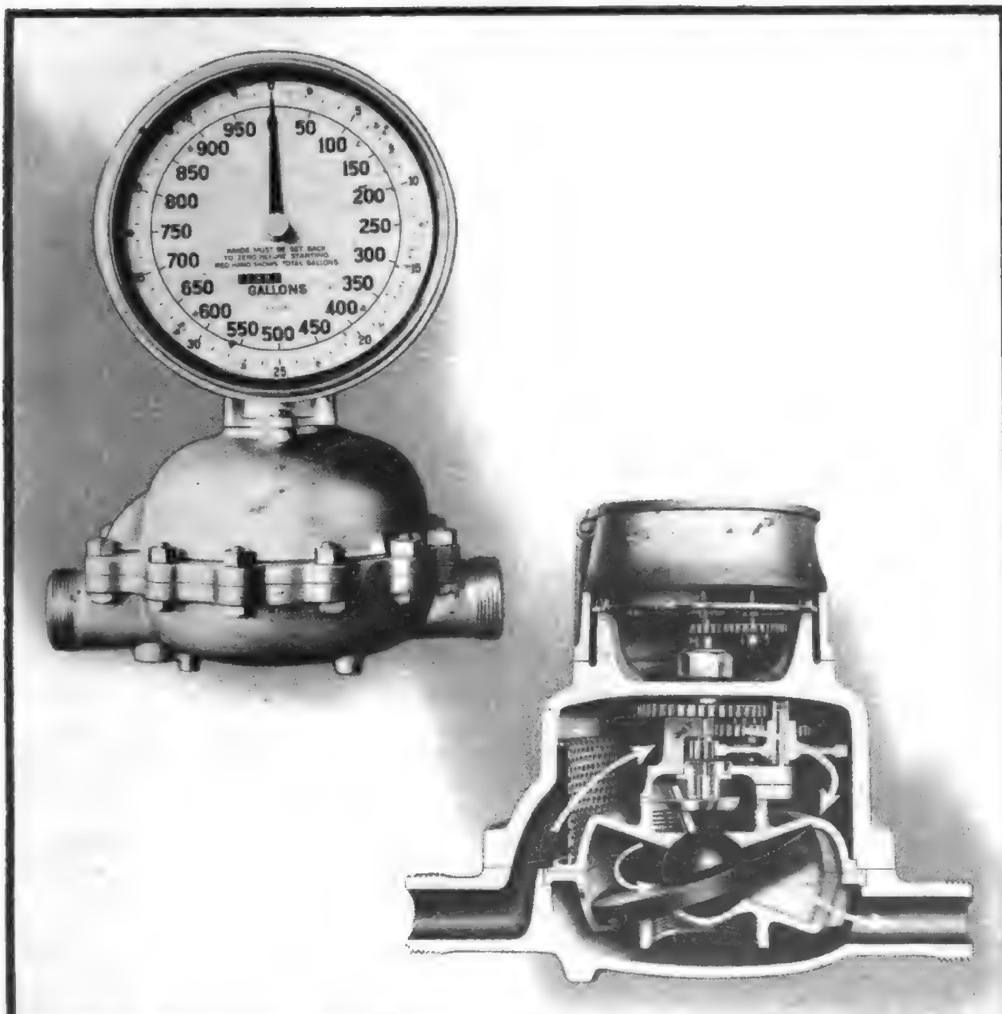

Figure 50

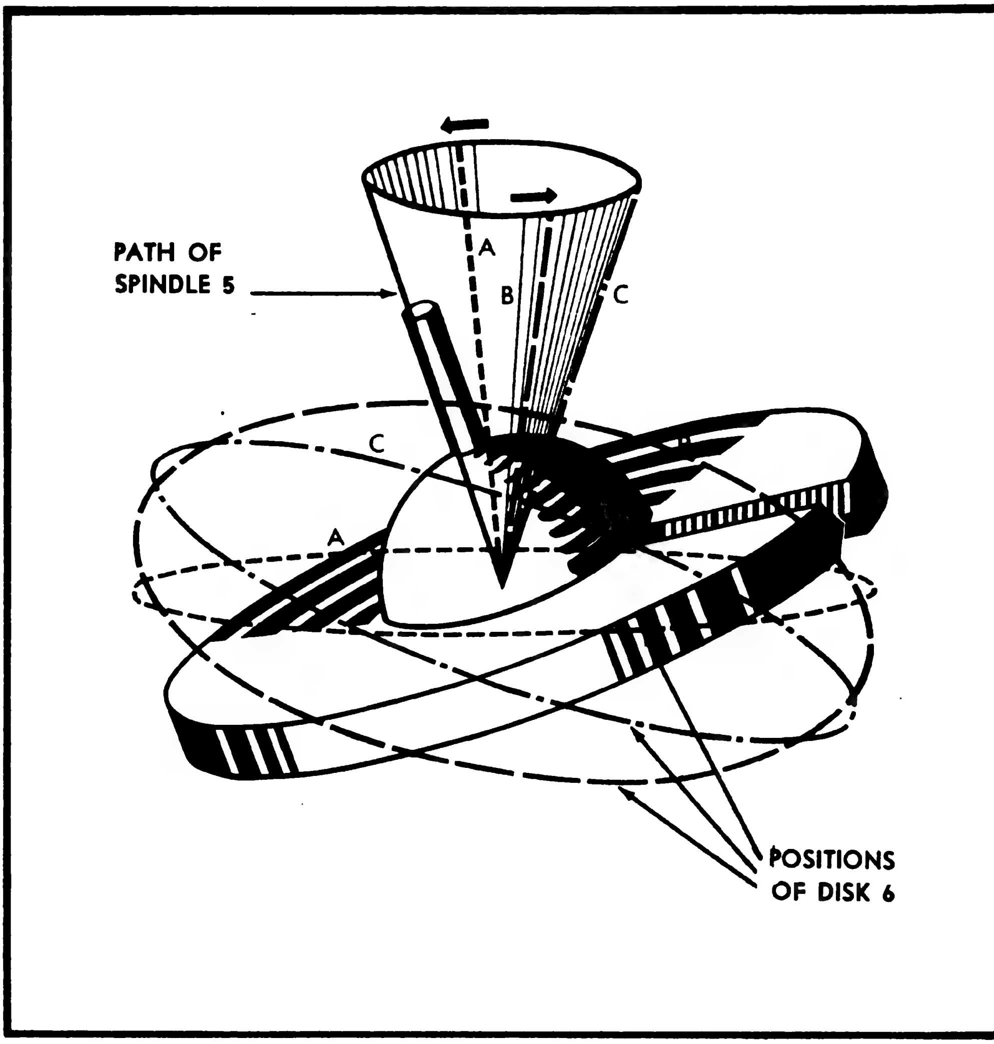

Figure 51

also be in position *A*, while positions *B* and *C* for the spindle and disk would also correspond.

But the disk cannot rotate, because partition *7* stands in slot *8*. The disk therefore wobbles up and down, while the seal lines formed by the disk and the top and bottom walls of the measuring chamber are made to revolve around the chamber.

For the sake of simplicity let us consider only the lower portion of the chamber. The seal will always be along the line which has the greatest inclination. As this seal line moves in the direction shown by the arrows in Figure 48, it will sweep the liquid before it and

cause it to be discharged through the opening *10.* At the same time it will be opening up an increasing volume in the compartment behind the moving line of the seal. Since this space is open to the inlet *9,* it will be filled with fresh liquid. When the line of seal passes the partition *7,* all the liquid formerly in front of the seal will have been forced out of the discharge. The seal line then starts to push the liquid which was formerly behind it forward. The line of flow of the liquid is as shown in Figure 50.

Obviously, if the wheel *11* were continuously rotated, the disk *6* would move a volume of liquid equal to the volume of the lower half of the chamber, from the inlet to the outlet, for every revolution. While this was going on, the top section would be doing the same thing, except that its seal is always directly opposite the lower seal in the measuring chamber. Therefore, for every revolution the piston will displace the volume of the whole chamber just once.

But we have been discussing the meter as though it were a pump operated by the rotation of wheel *11.* Actually the case is just the reverse. The meter is operated by the slightly greater liquid pressure at the inlet as compared with the outlet. This pressure difference causes the seal line to advance around the measuring chamber, and in doing so it rotates the wheel *11.* This in turn revolves the indicating hands on the face of the meter by means of shaft *12* and suitable reduction gearing.

Standard meters are suitable for temperatures to 200° *F*, and for pressures to 150 pounds per square inch, although models can be supplied for higher temperatures and pressures. The meters are accurate to about 1 per cent or less. The accuracy of the meter is not affected by pressure variations.

Cleaning. To clean the Niagara meter, unbolt the main flange and lift out the interior works for washing in kerosene or water. The strainer should also be thoroughly cleaned. Be sure to notice just where and how each part fits before removing it. The parts should not be forced, since they will go together easily if they are in the proper position. After reassembly the meter should be tested for

registry and smooth running, care being taken not to measure air or vapor rather than liquid.

To give accurate measurements a meter should be used only on the liquid for which it was designed; the strainer should be cleaned periodically of all sediment and pipe scale; and air should not be allowed to pass through the meter. To keep air out, do not let the meter drain between periods of use, so that the pump stuffing boxes and suction will be kept tight.

QUESTIONS

1. Describe the construction and operation of the Bourdon pressure gauge.
2. How can the Bourdon gauge be calibrated?
3. Describe the construction and operation of the Schrader pressure gauge.
4. How can the Schrader gauge be calibrated?
5. What precautions should be taken in the use and care of pressure gauges?
6. What determines the proper size of gauge to use in a hydraulic system?
7. Describe the construction and operation of the Niagara fluid meter.

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Chapter 4

PIPES, FITTINGS AND SEALS

This chapter deals with the pipes and tubes used for conveying liquid under pressure, and with the various devices employed to join units of hydraulic systems. It concludes with the Bureau of Ordnance requirements for the maintenance of pipes, tubes and fittings installed in ordnance hydraulic equipment.

Pipes

Materials used. From early times pipes were employed to convey water from one place to another. In ancient water supply systems, pipes made by binding four wooden slabs together in the general form of a square gave fairly satisfactory service, since the pressures were relatively low. With higher pressures, however, stronger materials became indispensable.

The first metal piping was crudely made from wrought iron, which contains very little carbon, a substance which makes iron brittle. Wrought iron is easier to work than steel with its larger carbon content, or than cast iron, which contains still greater quantities of carbon.

Wrought iron is still used for piping, since it is nearly as resistant to corrosion by acids as copper, which is a highly acid resistant metal. In addition, wrought iron is much cheaper than copper. Pipes and tubes are now also made from cast iron, steel, copper,

brass, tin, lead, aluminum, concrete, etc. On board ship, where piping is used for high-pressure water and high-pressure steam, low-pressure steam, fresh and salt water, oil, compressed air and refrigerating gases, practically all pipes and tubes are made of steel, copper or brass. In ordnance systems tubing made of copper or steel is used almost exclusively.

Steel pipe is relatively inexpensive, and is extensively used aboard ship because of its strength, its suitability for bending and flanging, and its adaptability to high pressures and temperatures. Its chief disadvantage is its comparatively low resistance to corrosion.

Copper piping and tubing is sometimes used on high-pressure lines, but is unsatisfactory for high temperatures or if subjected to repeated stresses, since it has a tendency to harden and break under these conditions. Thus at 360° F, the resisting strength of a copper tube is reduced about 15 per cent. Copper can easily be drawn out or bent, however.

Piping made of brass, an alloy of copper and zinc, is sometimes used instead of copper where the maximum working pressure is under 250 pounds per square inch. Brass is very resistant to corrosion, but cannot be bent as easily as copper. Brass piping is usually employed in low-pressure systems where threaded connections can be used, as for example in drain or lubricating lines.

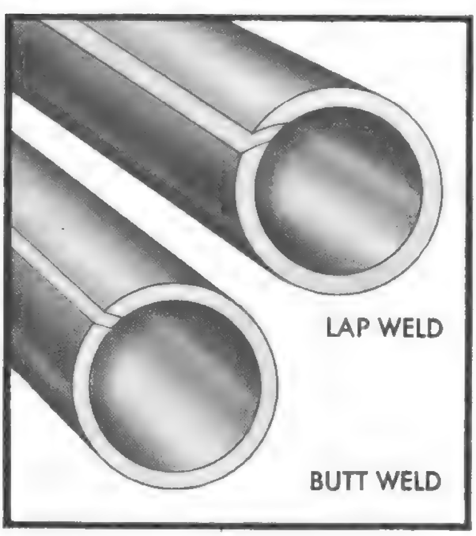


Figure 52

Methods of manufacture. Pipes and tubes can either be butt-welded or lap-welded, as shown in Figure 52, or hot metal bars can be pierced, drawn out and rolled into seamless pipe. Copper pipe comes either seamless or welded, while brass pipe is almost invariably seamless.

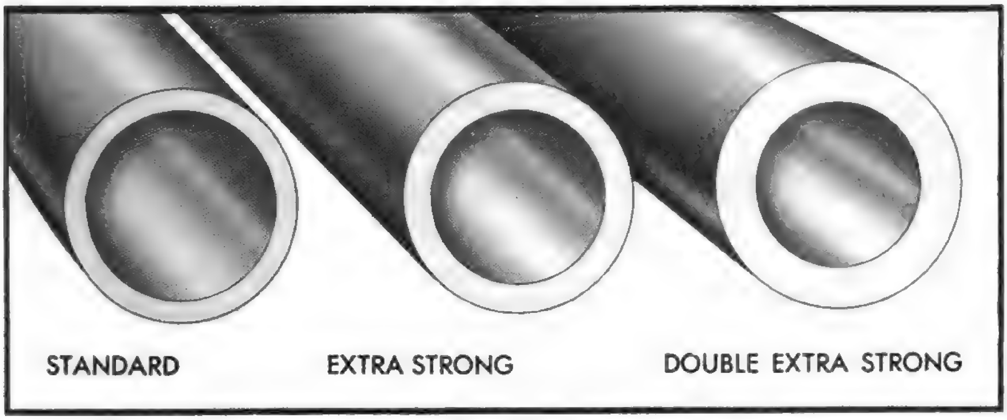


Figure 53

Size and thickness. Pipes are made with walls of three thicknesses—standard, extra strong and double extra strong (Figure 53). The exact thickness indicated by these terms depends on the size of the pipe. Standard thickness pipes up to 12 inches in diameter are referred to by an indicated or nominal inside diameter, but actual inside diameters vary considerably from the indicated figures. The situation is further complicated by the fact that extra strong and double extra strong pipes have the same outside diameters as standard pipes of the same indicated size. The walls are made thicker at the expense of the inside diameter. Above 12 inches, pipes are designated by outside diameter, as is the case also with copper and brass tubing of all sizes. Pipe is sometimes described in terms of its actual outside diameter and its thickness, as $6\frac{5}{8}''$ O.D. $\times$ 0.28″ wall.

Choice of materials and sizes. The demands that a hydraulic system must satisfy are often varied and complex, so that many different kinds of pipes, tubes and fittings must enter into consideration in designing an installation. The material to be used depends on the particular situation. Steel is employed where strength is required, or where high temperatures will be encountered. Copper or brass can be used where operating temperatures are below 350° *F*. In addition these metals are less liable to corrosion and offer less resistance to flow. Steel stands up better under vibration, however.

The maximum working pressure governs the material and the pipe

thickness to be used. Where resistance to high pressures is a primary consideration, seamless steel piping should be used. If the strength of a given seamless steel pipe be taken as 100, a lap-welded pipe of the same size will have a strength of about 92, and a butt-welded pipe a strength of about 73. Standard thickness 1-inch copper tubing will take pressures of over 1000 psi, while 2-inch tubes will take over 800 psi. For greater pressures, thicker tubes or pipes must be used.

The volume of flow to be carried determines the size of pipe to be used. Pipes should be large enough to conduct the maximum volume of flow at a moderate velocity, so that losses due to friction will not be excessive. Velocities of five feet per second are common.

Minimizing friction. The units of a hydraulic system are designed as compactly as is practicable, in order to keep the connecting lines short. Every section of pipe should be anchored securely in one or more places so that neither the weight of the piping nor the effects of vibration is carried on the joints. The aim should be to minimize stress throughout.

It often becomes necessary to introduce bends in the piping of a hydraulic system in order to reduce the number of joints or avoid obstructions that would otherwise require the use of several pieces of pipe and fittings. At the same time, however, bends should be avoided where possible.

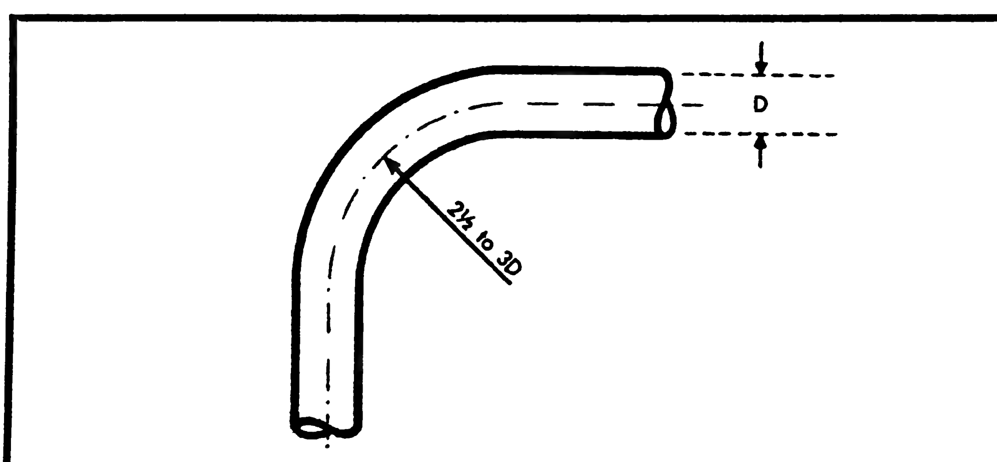


Figure 54

We have already pointed out that bends are described in terms of the ratio of the radius of the bend to the inside diameter of the pipe. The best radius for a pipe bend is between 2.5 and 3.0 times the inside pipe diameter, as shown in Figure 54. If the inside diameter of a pipe is 2 inches, the radius of the pipe bend should be between 5 and 6 inches.

While friction head increases markedly for sharper curves than this, it also tends to increase up to a certain point for gentler curves. The increase in friction in a bend with a radius of more than about 3 pipe diameters comes from increased turbulence on the outside edges of the flow. Particles of liquid must travel a longer distance in making the change in direction. When the radius of the bend is less than about 2.5 pipe diameters, the increased pressure loss is due to the abrupt change in the direction of flow, especially for particles on the inside edge of the flow.

Fittings: Screwed Fittings

Varieties and use. Fittings are used to connect the units of a hydraulic system, including individual sections of pipe. The pipe fittings most commonly used in ordnance hydraulics are either screwed or flanged, while for tubing the fittings are either flanged or of compression type. Each kind of fitting is used for a particular purpose.

Screwed fittings are commonly employed in low-pressure pipes such as lubricating or drain lines. In Navy systems they are usually made of steel, copper or brass, and in a variety of designs of which a few are shown in Figure 55. Screwed fittings are made with standard female threading—threading, that is, cut on the inside surface—and require that the end of the pipe be threaded with outside male threads for connecting.

Threading pipe. The threads should be cut clean and smooth. The dies and pipe cutting tools should be sharp so that they will not drag metal. Otherwise threads will fit poorly, and leakage may result. Use oil liberally when cutting threads. Standard pipe threads are tapered slightly to insure tight connections. The amount of taper is

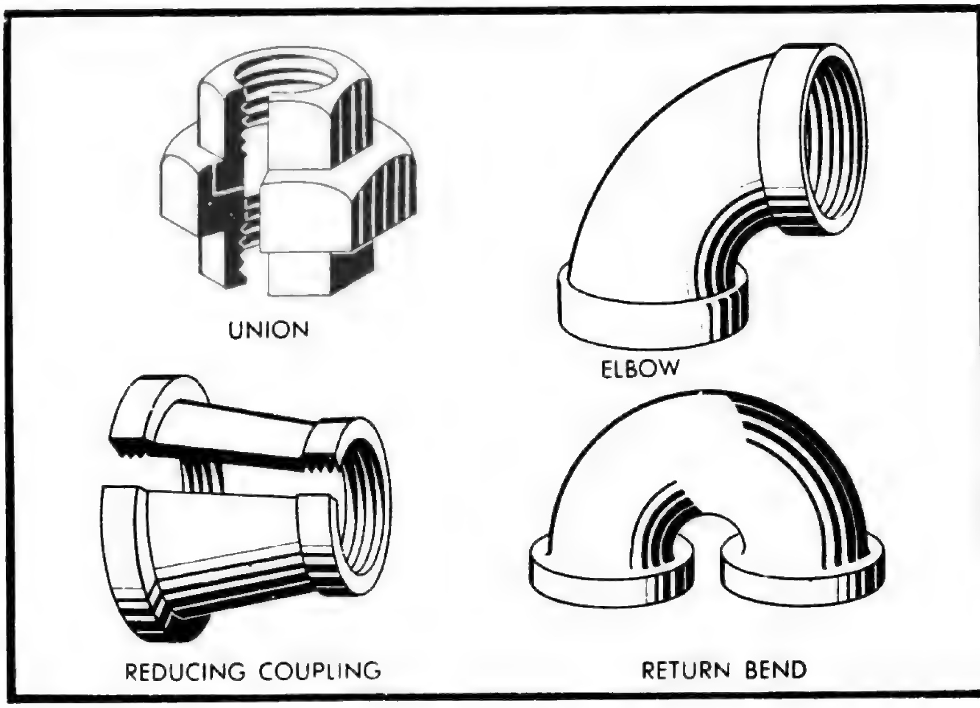


Figure 55

about $\frac{3}{4}$ inch in diameter per foot of thread (Figure 56).

After pipe has been cut or threaded, the ends should be inspected for burrs. All projections and sharp edges, inside and out, should be removed (Figure 57). If this is not done, they will obstruct the flow of liquid, and loose particles of metal may get into the pipe line and damage the system.

Metal is removed when a pipe is threaded, thinning the pipe and

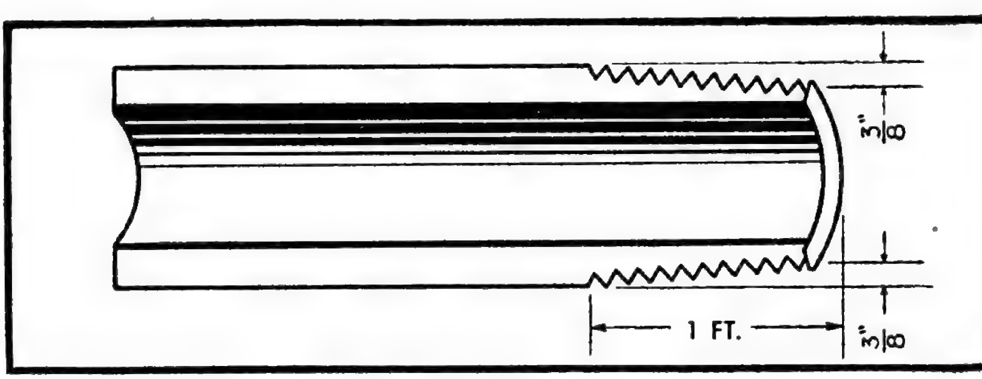


Figure 56

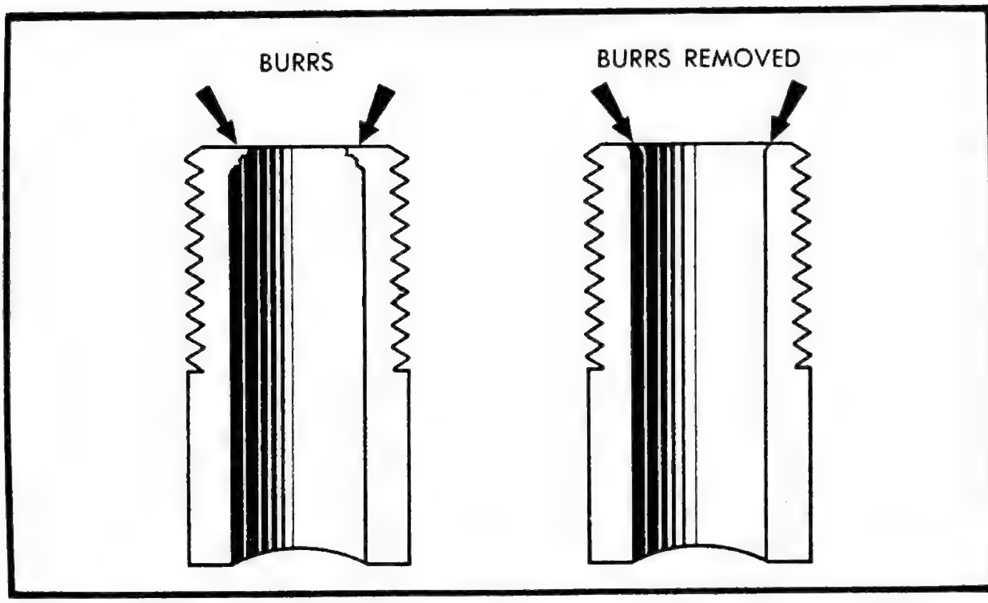


Figure 57

exposing new and rough surfaces for chemical action. Corrosive agents work more quickly at such points than elsewhere. If pipes are assembled with no protective compound on the threads, corrosion sets in at once and the two sections stick together so that the threads seize when disassembly is attempted. The result is damaged threads and pipes.

To prevent seizing, permatex or some similar "pipe dope" should be placed on the threaded portion of the pipe down to about the third thread from the end (Figure 58). The anti-seize compound should not be placed at the extreme end of the threading, or some particles may loosen into the pipe line and cause unnecessary damage in the system. It is placed on male fittings only, so that the screwing process will not push the compound into the system. To assure a good fit free from leakage, at least two-thirds of the threaded portion of the pipe should be screwed up into the fitting. The connection should be made tight with a wrench.

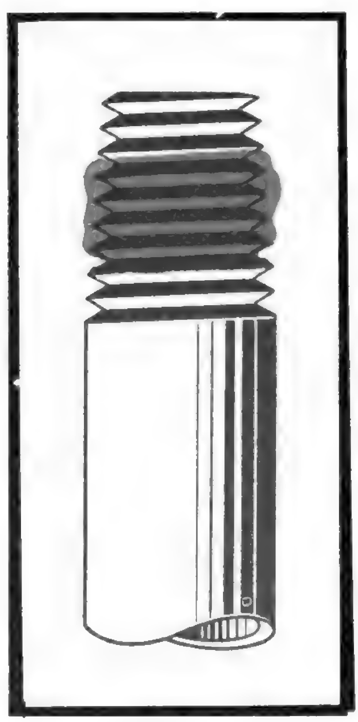


Figure 58

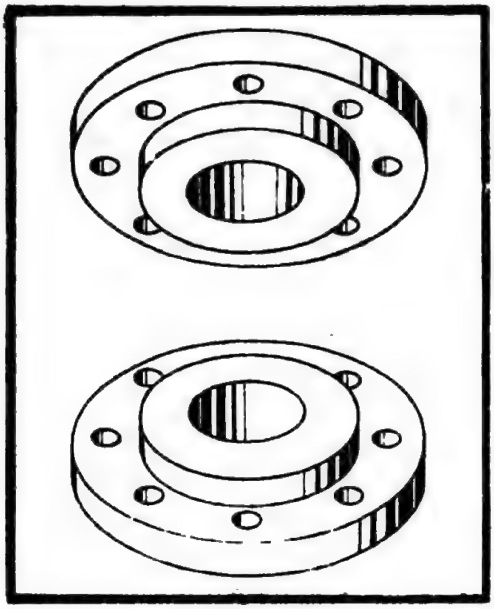


Figure 59

Flanged Fittings

Varieties and use. Flanges or offsets of many different kinds can be fitted to pipe ends in various ways so as to hold them together. They are commonly used on high-pressure lines and large diameter pipes. The most usual type consists of two cast or forged flanges (Figure 59), secured to the pipe ends and bolted together as shown in Figure 60. A gasket is placed between the meeting faces of flanges in order to make a tighter connection.

In hydraulic installations, flanges are either screwed to the threaded end of the pipe and backwelded, as in the cross section view of Figure 61; or they are slipped over the end of the pipe, welded front and back, and then refaced flush with the pipe end (Figure 62). The welding eliminates any possibility of leakage between the flange and the pipe.

Types of flange face. The joining surfaces of flanges can be faced in many ways to secure a tight joint. Figure 63 shows three of the most common types of faces used in hydraulic installations.

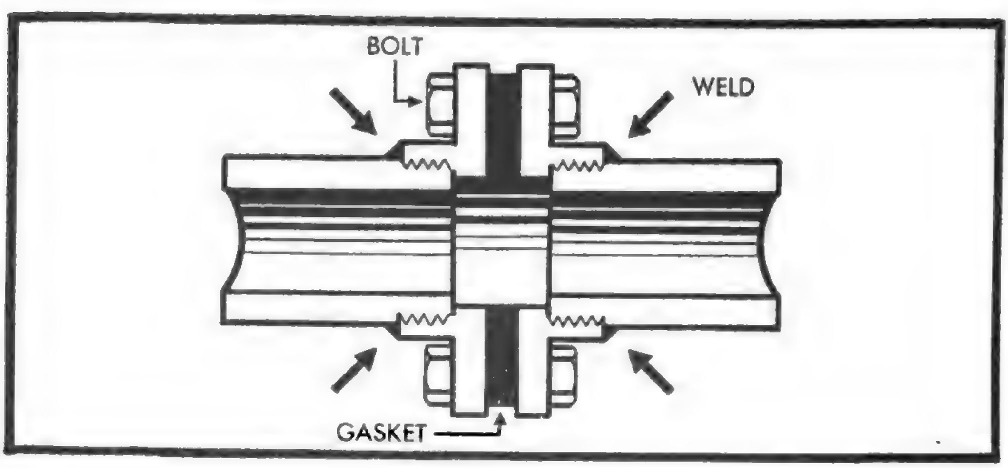
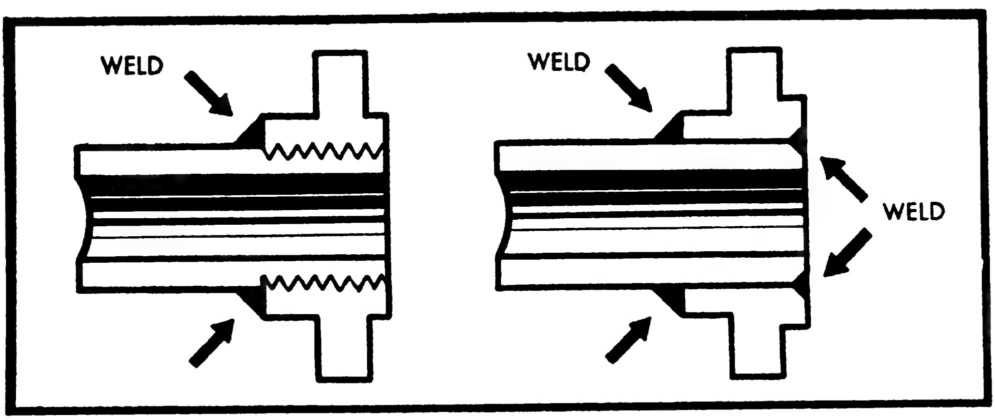


Figure 60

Flanges with plain raised faces have a machine ground surface, and may be used with either a square or ring gasket. With this type of joint the flanges must meet perfectly. Yet the plain raised face type has several advantages over other types, in that all mating of flanges is eliminated, the gasket is easy to center, and a section of piping fitted with this kind of flange can easily be removed without springing the line apart.

The male and female type of flange face has a cut away (female) area on the face of one flange and a corresponding raised (male) area on the face of its mate. It normally takes a small ring gasket. This makes a more secure joint than the plain raised face type. The tongue and groove type of face has a raised ring on one flange and a corresponding ringed groove or cut away area on the other flange.



Figures 61 and 62

Flanges that must be mated are undesirable from the maintenance standpoint, since interconnected units have to be sprung apart in order to be removed from the line. They are used, however, in high-pressure installations, where the plain raised face type does not make a satisfactory connection.

Flanged joints are usually bolted securely together by four bolts. The flanges must line up squarely. When being assembled, bolts diametrically opposite should be set up fairly tight. Then the other two bolts should be set up an equal amount. The bolts should then be taken in order, setting up each one an equal amount, until on the last round they are all equally tight.

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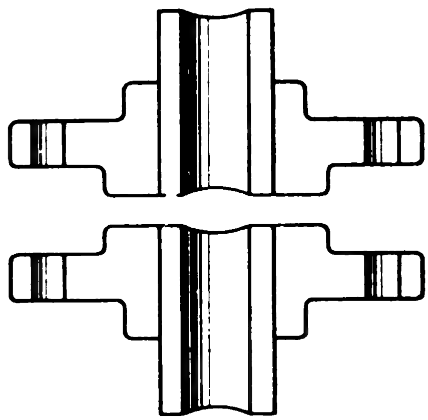
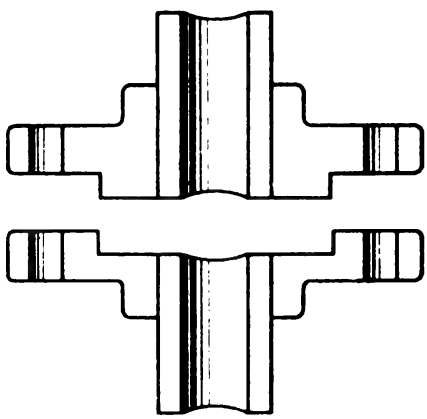

PLAIN RAISED FACE


MALE AND FEMALE

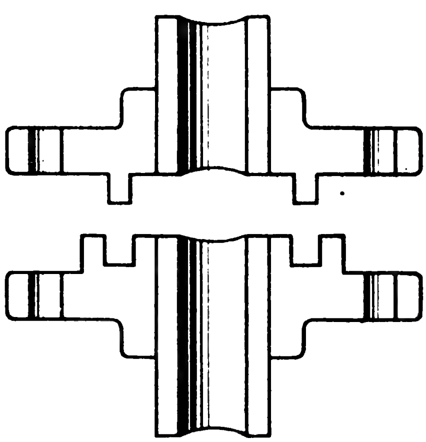

TONGUE AND GROOVE

Figure 63

The flanged fittings used on hydraulic tubing are identical with flanged pipe fittings. They are used on high-pressure tube lines.

Compression Fittings

Compression fittings are used in medium or low-pressure tube lines, and are of many kinds. Some of those most commonly used in ordnance hydraulic equipment are shown in Figure 64.

The Bell type compression fitting consists of a tube nut and a tube coupling so threaded that the coupling will screw down over the nut. The nut is fitted over the tube, and the end of the tube is then flared. The nut and the flared tube end are then fitted into the female opening of the tube coupling (Figure 65). As the tube nut is tightened, the flared end of the tube is compressed against a machined taper in the tube coupling, providing a firm connection.

The Parker triple compression fitting is very similar, except that a sleeve is placed between the flared tube and the nut, as shown in Figure 66. The sleeve protects the flare from strain as the nut is tightened, since the nut turns against the shoulder of the

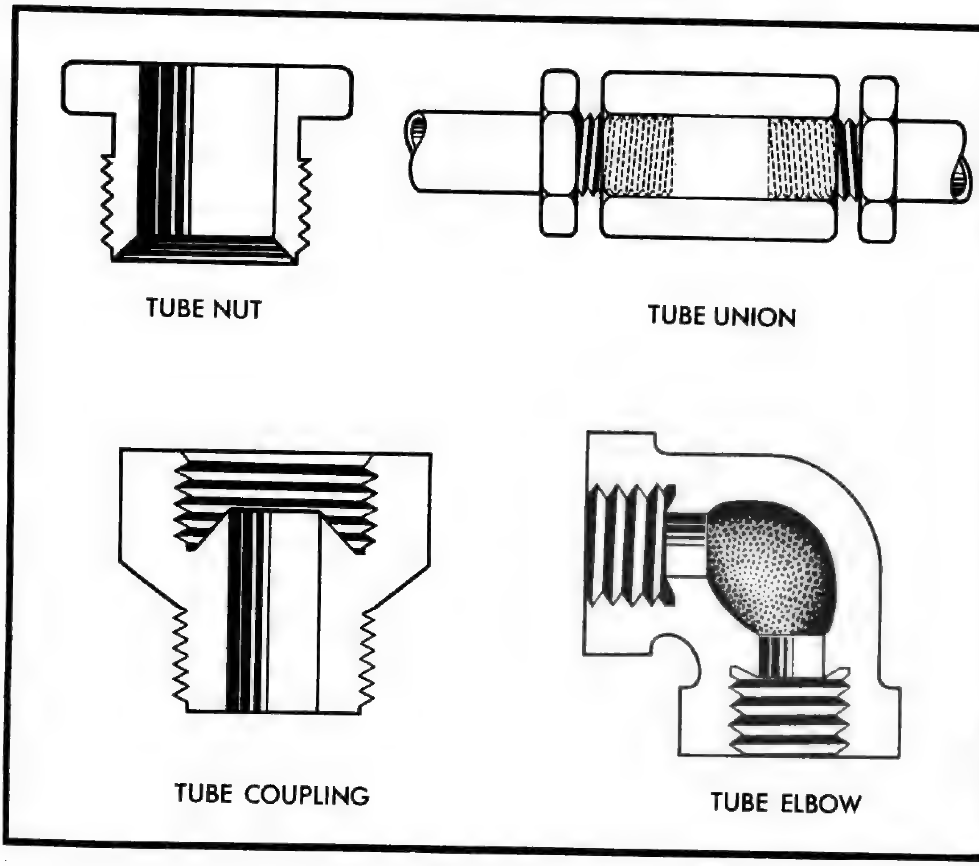


Figure 64

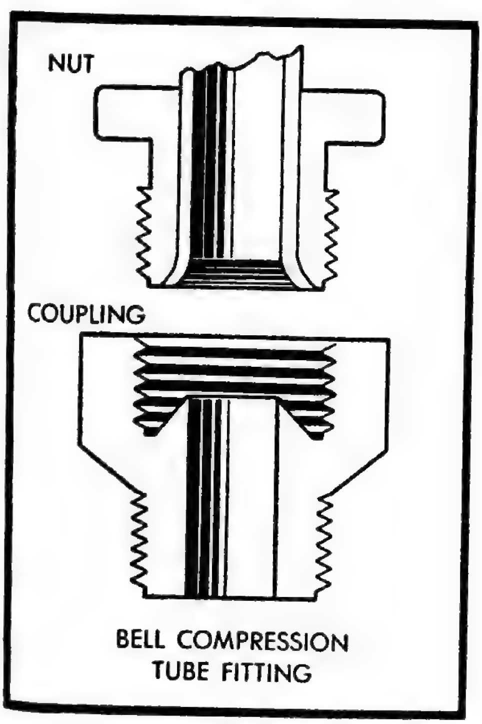


Figure 65

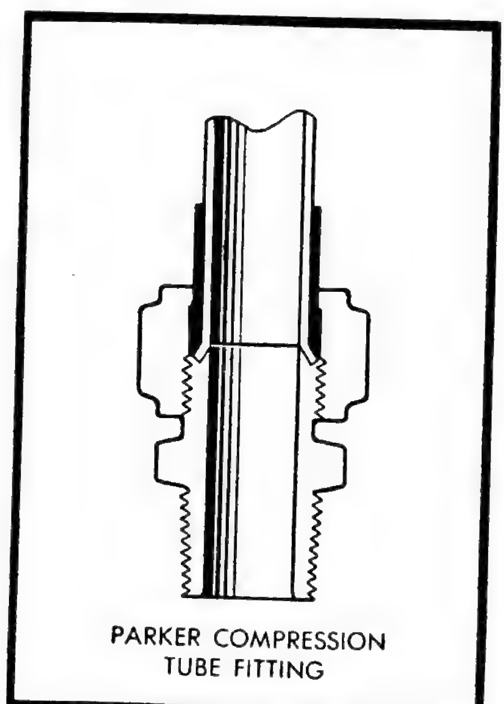


Figure 66

sleeve instead of directly against the flare.

Flaring. In order to obtain a firm compression connection the end of the tube must be flared properly. A hand flaring tool is shown in Figure 67. The flare is made by slipping the tube nut over the end of the tube, inserting the rounded end of the flaring tool into the tube, and rolling the side of the tool over the end of the tube to flare the tube against the nut.

The seat of a good flare is smooth and even, with the flared end projecting just beyond the end of the tube nut. The maximum diameter of the flare should be slightly less than the nose of the tube nut. In Figure 67, a good flare is shown, along with two badly made flares.

When the tube is properly flared and the tube nut tightened, the flare will seat itself firmly in the machined recess of the tube coupling. The tube will not turn relative to the coupling when the nut is tightened, and the flare will assume the shape of the machined surface. Any irregularities in the machined recess will be taken up by the tube flare, making a leak-proof connection.

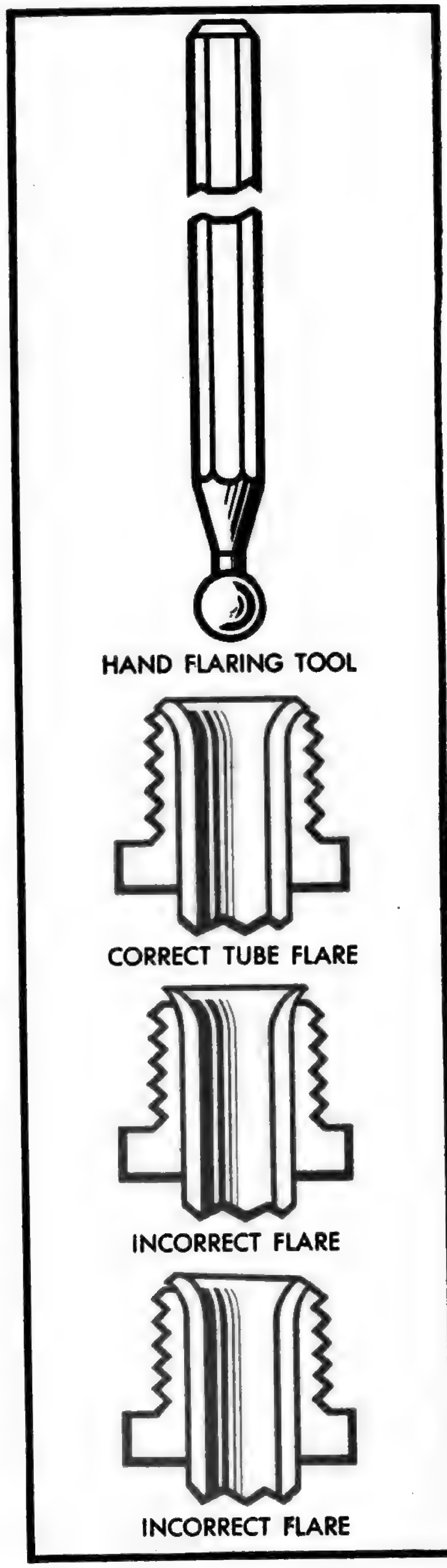


Figure 67

If the original connection is broken, however, and the compression tube fitting must be reassembled, it is very improbable that the flare will contact the machined recess in the original places. It is therefore necessary to anneal or soften the flared end of the tube prior to reassembly.

Copper can be annealed by heating it to a red heat and then quickly quenching it in cold water. This process will soften the copper enough to allow it to assume the shape of the coupling surface when the tube nut is tightened.

Only copper tubing is to be annealed. When compression fittings are used with steel tubing, a thin copper gasket is placed as a seal between the tube flare and the machined recess. This copper gasket must then be softened before reuse.

Hydrostatic test before use. All piping, tubing and fittings used in hydraulic systems must be subjected to a hydrostatic or static pressure test before being put to use. The system is filled with water and subjected to a test pressure of approximately 50 per cent more than the maximum planned working pressure for each part. In this way a system free from leaks and possible breaks will be assured.

Gaskets, Oil Seals and Packing Rings

Gaskets made of relatively soft materials are placed between the meeting surfaces of hydraulic fittings in order to increase the tightness of the seal. They should be composed of materials that will not be affected by the liquid to be enclosed, nor by the conditions under which the system operates, including maximum pressure and temperature. They should be able to maintain the amount of clearance required between meeting surfaces.

The selection of correct seals and their proper installation is an important factor in maintaining an efficient hydraulic system. Commercial manufacturers specify the kind of gaskets to be used on their equipment, and their instructions are to be followed in re-

placing gaskets. If the proper gasket is not available, careful consideration should be given to the selection of a suitable substitute.

Neoprene gaskets. One of the materials most widely used in making gaskets for naval ordnance hydraulic systems is neoprene, a synthetic rubber and fibre composition which comes in sheet form to be used between flat surfaces and in molded packing rings to be used around movable rods and shafts. Since neoprene is flexible, it is often used in sheet form at points where a gasket must expand enough to allow air to accumulate, as with cover plates on supply tanks, etc.

Over a period of time oil tends to deteriorate the material used in making neoprene gaskets. For this or mechanical reasons, small particles of the material may be cast free into the system, and may plug small orifices and restricted passages. The condition of the gasket must therefore be checked whenever the unit is disassembled.

Since neoprene gasket material is soft and flexible, it easily becomes misshapen, scratched or torn. Great care is therefore necessary in handling neoprene. Shellac, gasket sealing compounds or "pipe dope" should never be used with sheet neoprene, unless absolutely necessary for satisfactory installation.

Oil-treated paper. This widely used gasket material is manufactured by a number of companies under different trade names—as Gaskoid, vellumoid, consolco. Gasket paper usually comes in sheets, and the repairman must cut the gasket for each particular installation. The paper should be marked to size and cut with a sharp knife or snips. It should not be placed on the valve plate or other surfaces and tapped with a ball peen hammer to make the desired passages, since this will cause burrs.

Sheets of gasket paper vary in thickness from several thousandths of an inch to almost any desired thickness. A thick gasket is not always the best. Where the gasket is to be installed between a valve plate and the valve block, for example, the thickness of the gasket may determine the clearance of working parts.

When oil treated paper gaskets are torn, wrinkled or bent, leaky connections may result. If gasket shellac or cement is used, it should be applied to one side only so that the gasket will not be torn when disassembly is attempted.

Lead foil. In some high-pressure connections with only slight clearance—usually from 0.001 to 0.005 inch—lead foil is used as gasket material. It is soft and pliable, so that it easily fits an uneven contour. When disassembling a unit where lead foil gaskets have been used, great care is necessary not to bend or tear the foil if it is to be used again.

Cork. This material makes a good oil seal and can be used under various pressure conditions. Cork is in fact one of the best gasket materials available. It is very compressible and can be cut to any desired thickness to fit any surface, and yet can be made to give a desired clearance while forming an excellent oil seal.

Cork requires the same care in disassembly as any other soft gasket material. The condition of cork gaskets should be checked for defects when a unit is disassembled, so that small particles will not be cast off into the system during use.

Copper. This material is used extensively in naval ordnance hydraulic systems, both in the form of flat copper rings used under adjusting screws, etc., to give oil seals, and as molded copper rings

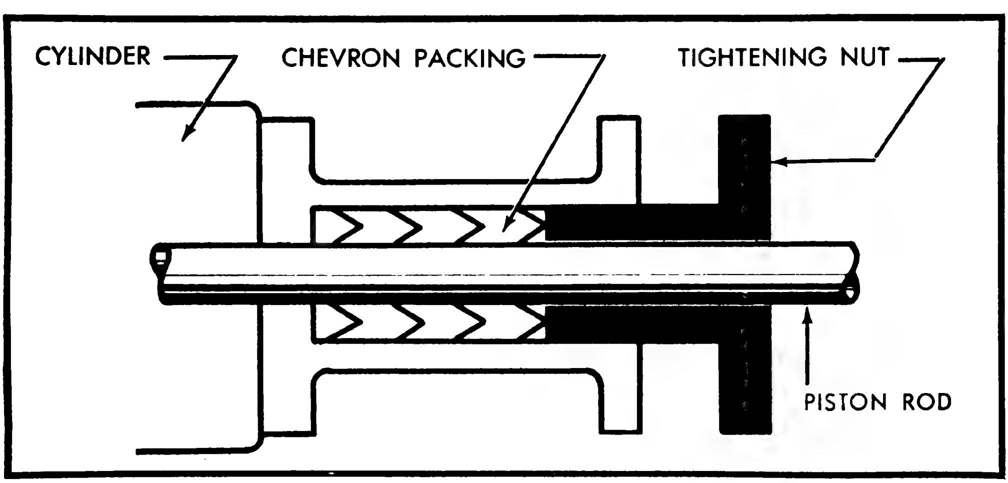


Figure 68

to be used with speed gears, etc., operating under high pressures. Either type is easily bent and requires careful handling. In addition copper becomes hard when used over long periods or if subjected to compression. Whenever a unit is disassembled, the copper gasket rings should be softened by annealing.

Neoprene packing rings for moving parts. The neoprene used for chevron packing rings is made of a harder composition than that used for gaskets, and is molded to fit a particular stuffing box and rod. This type of ring is sometimes filled with soft neoprene strips. Under pressure the chevron packing rings are forced together so that they spread out, at the same time causing the soft filler to expand and thus force the feathered chevron edges more firmly against the rod. Figure 68 shows a number of chevron packing rings in place around a piston rod.

Before installing this type of ring, it should be soaked in oil for several hours to help it assume the shape of the rod. If it is not possible to soak the packing, the tightening nut should not be tightened enough to bind the rings or cause them to become misshapen.

Until the packing becomes set, the leakage around the shaft may seem excessive. Since a small amount of leakage is unavoidable, and is necessary for lubrication of the rings and shaft, the tightening of the retaining element is an operation requiring great care.

When disassembling a hydraulic unit where this type of packing ring must be removed over a threaded shaft or rod, the threads should be covered with a thin piece of sheet brass or a cap usually provided for this purpose, in order that the threads will not damage the interior of the packing ring.

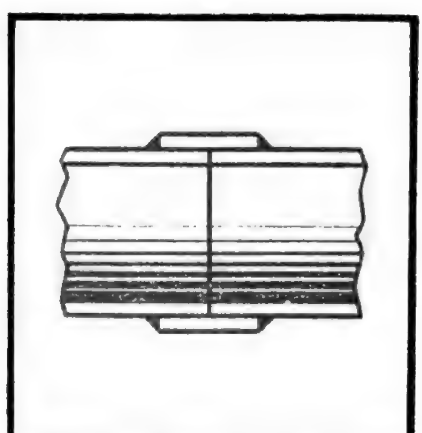


Figure 69

Steel sealing sleeves. A sleeve joint or coupling is sometimes used to

connect lengths of pipe in a high-pressure line. It consists of a short steel cylinder, from 0.10 to 0.15 inch larger in diameter than the pieces to be joined. The sleeve is fitted over one of the pipe ends, the other pipe end is introduced, and the two ends carefully aligned (Figure 69). The two ends of the sleeve are set in place and flange-held, or they may be welded to the pipe. The connection must be drawn-up evenly, since a sleeve out of alignment will almost invariably leak.

Maintenance of Hydraulic Equipment

General requirements. When installing the various pipes, tubes and fittings of a hydraulic system, it is necesary that they be *absolutely clean, free from scale, and from all kinds of foreign matter.* The Bureau of Ordnance sets forth strict requirements for pipes, tubes, or fittings installed on ordnance hydraulic equipment. The general requirements are as follows:

1. All pipe, tubes, and fittings must be absolutely clean and free from scale and all foreign matter.

2. All pipe, tubing and fittings on which work has been done shall be cleaned in the manner shortly to be described, and protected after all work operations are completed.

3. The interior surfaces shall be treated to prevent rust or corrosion and all openings shall be sealed or plugged to exclude foreign matter during shipment and storage prior to installation.

4. All pipe, tubing and fittings shall be recleaned and repickled whenever heat is applied for the purpose of bending, annealing, or performing other operations after removal of the original scale.

5. All pipe, tubing, and fittings shall be thoroughly cleaned, just prior to installation.

Safety precautions. Dangerous chemicals are used in the operations to be described. They should be kept only in the proper containers and *handled with extreme care.* Many acids give off dan-

gerous fumes when brought into contact with air, and can injure clothing or human tissue even when in the gaseous state. Contact with these chemicals or their fumes should be avoided. The use of rubber gloves and aprons is advisable, and special acid goggles should be worn if extended operations with acids are necessary.

Hydrofluoric acid must be kept in lead, paraffin or rubber containers, since it very quickly decomposes glass. In mixing acids with water, especially in the case of sulphuric acid, the acid should be slowly added to the water, and not the opposite, or a violent explosion may result. The point is to begin with a very dilute solution and gradually increase its strength.

An alkali like lye is also capable of injuring clothing and tissue if present in sufficient concentration. The best procedure, of course, is to avoid direct contact. None of these chemicals should be taken internally under any circumstances. If contact with any of them does occur, and the resulting wound is anything but trivial, medical aid should be summoned at once, the affected parts should be liberally doused with cold water, and steps taken to neutralize the chemical action. This means that acids should be treated with *dilute* solutions of *weak* alkalis, and alkalis with *dilute* solutions of *weak* acids. A very dilute solution of ammonia can be applied to an acid burn, and vinegar, which is a dilution of acetic acid, can be further diluted and applied to alkali wounds. These treatments are not advisable if the eyes have been affected, except for the dousing with water. A boric acid solution, however, can be applied to the eyes. After treatment the wounds can be covered with an oil dressing.

Procedure for copper and brass. The following procedure for cleaning and pickling copper or brass pipe, tubes, and fittings originated at the Naval Gun Factory and, after use over a period or several years, is approved and recommended:

1. <u>Preparation for pickling to remove scale:</u>

   a. Brush with boiler tube wire brush, or clean with commercial pipe cleaning apparatus, if necessary.

b. Clean the piping or tubing in lye solution (6 oz per gal) at 160 to 180° *F* to remove oil or grease, if present.

c. Rinse with hot water.

2. The pickling operation:

a. Pickle free from scale in a hot solution (110 to 140° *F*) of the following composition:

Substance	*By weight (Ounces)*	*By volume (Pint)*
Commercial sulphuric acid (Sp. Gr. 1.84)	8	0.28
Commercial hydrofluoric acid, 30 per cent	17.8	0.96
Water	112.2	6.75
Total volume	138.0 oz =	1 gallon
	8 lb 10 oz	

The tubing should be pickled a sufficient length of time to remove scale. The position of the tubing or piping in the pickling bath should be changed occasionally to make sure that if gas pockets form, they will not always be at the same location. Precautions should be taken in pickling brass or copper pipe or tubing with steel fittings. Ordinarily the pipe or tubing will be pickled clean before the steel fittings are attached, but if heavy scale is encountered, it may be necessary to protect the steel fittings with paraffin during pickling.

b. Rinse with water after the scale has been removed and bright dip by pouring a solution of the following composition through the tubing to remove residue from the pickling operation and brighten the surface:

Substance	*By weight (Ounces)*	*By volume (Quart)*
Commercial sulphuric acid (Sp. Gr. 1.84)	107.4	1.75
Commercial nitric acid (Sp. Gr. 1.42)	13.4	0.28
Commercial hydrochloric acid (Sp. Gr. 1.18)	0.34	*
Water	65.26	1.96
Total volume	186.40 oz =	1 gallon
	11 lb 10.4 oz	

* 0.25 Fluid ounce.

3. Treatment subsequent to pickling operation:

   a. Rinse the acids remaining from the bright dip from the tubing with water. Immerse the tubing for 2–4 minutes in a solution containing 4–6 oz per gallon of lye or sodium carbonate to remove all traces of acid. Rinse with hot water.

   b. Immerse the tubing or piping in a solution containing 16 oz of chromic acid and 4 oz of sulphuric acid per gallon of water, to remove tarnish. Rinse with hot water and blow out with dry air.

   c. Use wire brush. Blow out thoroughly with dry air and inspect immediately for the presence of scale. Do not use rags or waste in any of the drying operations since lint or thread may be retained in the tubing.

   d. Seal all openings of the pipe, tubing and fittings to prevent the entrance of foreign matter.

4. Cleaning copper, brass or bronze pipe fittings prior to installation:

All pipe, tubing and fittings are to be cleaned, after all other work is finished, and just prior to installation of the piping. If for any reason the parts are not used for a month or more after being cleaned, they should be recleaned before being installed. The following procedure is recommended:

   a. Dip in acid bath consisting of the following and remove immediately:

      2 parts of sulphuric acid
      4 parts of water
      1 part of nitric acid

   b. Wash in cold water.

   c. Immerse for about one minute in Magnus No. 2 solution of sufficient concentration to neutralize all traces of acid from the pickling operation.

d. Wash in fresh cold water.

e. Soak in boiling hot water for 10 minutes.

f. Use wire brush or pass frayed wire, according to size of pipe, through the pipe.

g. Wash with stream of fresh cold water at high pressure.

h. Dry thoroughly with air. Do not use rags, waste or toweling in the drying process.

i. Seal all openings until ready to install.

In the case of machined faces, or threads on the fittings, treat these with paraffin wax before starting to prevent acid from acting on these surfaces.

Procedure for iron and steel. The following procedure is recommended for the pickling and cleaning of iron or steel pipes, tubes and fittings:

1. Preparation for pickling to remove scale:

   a. Grind out spatter from welding or brazing operations.

   b. Brush with a boiler tube brush. If this is not available for small tubing, use a brush improvised by cutting the out strands of a section of a steel wire cable and turning them up to form a brush. The diameter of the cable should be $\frac{1}{16}$ to $\frac{1}{8}$ inch less than the interior diameter of the tubing and the length of the cable should be slightly over twice the length of the longest tubing in which it is used. This preliminary treatment serves to remove loose particles and to indicate that the tubing is free from major obstructions.

   c. Immerse the tubing in a lye solution (6 oz per gal) at a temperature of 160–180° F for 4 hours to remove oil or grease previous to pickling.

   d. Rinse with hot water to remove the lye solution.

2. The pickling operation:

   a. Pickle to remove scale in a hot solution (110–140° *F*) of the following composition:

Substance	*By weight (Ounces)*	*By volume (Pint)*
Commercial sulphuric acid (Sp. Gr. 1.84)	14.3	0.5
Commercial hydrofluoric acid, 30 per cent	7.4	0.4
Commercial inhibitor	.067–.134	--
Water	---	7.1
Total volume		1 gallon

The time of pickling will be determined by the amount and depth of scale on the interior surfaces of the tubing. The position of the tubing during pickling should be changed occasionally to make sure that if gas pockets form, they will not always occur at the same location.

3. Treatment after pickling:

   a. After the scale has been removed, rinse the tubing first with hot water, then with cold water. Immerse the tubing in a solution of lye (6 oz per gal) from 5–10 minutes to remove all traces of acid.

   b. Rinse free from the lye solution with hot water and immerse the tubing in a solution containing 4 oz per gal of sodium cyanide for 2 hours. Rinse thoroughly with hot water. Blow out with dry air until the surfaces are dry.

   c. Use wire brush. Blow out with dry air under high pressure.

Do not use rags or waste in any of the drying operations since lint or thread may be retained in the tubing.

4. Inspection and protection of interior surfaces:

   a. Inspect the tubing for the presence of scale immediately after the final drying operation. Immediately after inspec-

tion, flush the interior of the tubing with rust preventive compound (52–C–18) and allow it to drain. The coating of light oil serves as a temporary protection against corrosion during the interval (1–3 days) between the pickling operation and the hydrostatic tests. After oiling, plug both ends of the tubing to exclude foreign matter and remove from the plating shop as soon as possible to avoid attack from fumes.

5. Treatment during hydrostatic tests:

   a. Remove the rust preventive compound on the interior surfaces of the tubing by dissolving in Stoddard solvent (Fed. Spec. P–S–661) previous to the hydrostatic tests.

   b. All water used in the hydrostatic tests must contain 0.66 oz of neutral CP sodium chromate to a gallon of water. The presence of the sodium chromate in the water retards the formation of rust. After the hydrostatic tests have been made, drain the solution from the tubing and blow out with dry air until the interior surfaces are dry.

   c. After the interior surfaces of the tubing are dry, apply rust preventive compound (52–C–18) and rub in thoroughly with a bristle brush.

The application of the rust preventive compound provides a protection against corrosion of the interior surfaces of the tubing during storage and shipment. After the rust preventive compound has been applied, plug both ends of the tubing with wooden plugs to exclude foreign matter during storage and shipment. The rust preventive compound should not be removed until just before installation. A preparation known as Stoddard solvent may be used to remove the rust preventive compound.

6. Conditions under which repickling is necessary:

   a. If the subsequent installation of the tubing involves the application of heat (such as in bending) the tubing should be

repickled according to the procedures already described since the application of heat is likely to cause the formation of scale on the interior surfaces of the tubing.

b. If repickling is necessary, the hydrostatic test should be repeated.

QUESTIONS

1. What are the most important characteristics of a good pipe system?

2. Explain the conditions determining what materials shall be used for piping or tubing in a hydraulic system.

3. How is pipe and tubing of different dimensions distinguished? What is the difference between standard and double extra strong pipe?

4. What is the difference between male and female threading?

5. What is meant by the radius of curvature of a pipe? What should be the radius of curvature of a pipe whose inside diameter is 3 inches?

6. How is permatex or other anti-seize compound applied?

7. Describe the various types of flanges and the conditions under which they are used.

8. Describe compression fittings, including flaring and annealing.

9. What is pickling?

10. How is pipe cleaned?

11. Why are gaskets used? Of what material are they composed?

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Chapter 5

SIMPLE VALVES

This chapter covers the following topics: what valves are, and how they are used; the classification of valves; different types of simple valves—cocks and gate, globe, needle, check, relief and safety valves; the installation and maintenance of valves.

What Valves Are, and How They Are Used

Definition and use. A valve consists of an opening in a pipe and some means of closing it. A surface may be pressed against the opening, as one closses a door, or a surface may be moved across the opening, as one closes a window (Figure 70). One of the two meeting surfaces must overlap the other, so that the resulting seating surface will give tight closure.

It is all but impossible to develop a practicable hydraulic system without using valves, and it is equally difficult to manage a hydraulic system without knowing how to control its valves and insure their efficient operation. They are used in hydraulic systems for three principal purposes—to control the volume or the direction of flow, or the pressure at which the system operates. They may be located either at exits from the system or within it. A typical example of a valve used to control volume of flow located at an exit is an ordinary faucet.

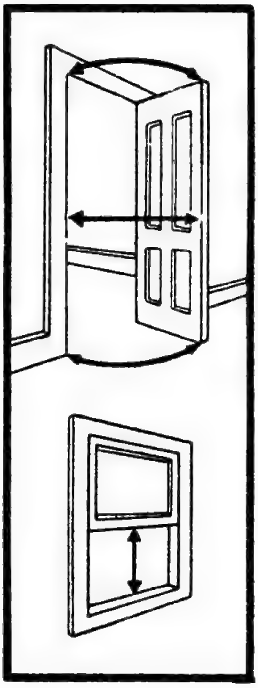


Figure 70

Historical. Even in early irrigation systems we find sluice gates—really crude valves—that could be raised or lowered to control volume of flow (Figure 71). By closing one such gate and opening another, the direction of flow could also be controlled. When the hydraulic press was invented in the eighteenth century, it utilized a manually operated gate valve very different in structure from the Egyptian sluice gate, but performing exactly the same function of cutting off or allowing flow. A gate valve was used in the hydraulic jack described in Chapter 1. When it is desired to lower the lift platform, this valve is opened and liquid in the lift cylinder flows back into the reserve tank. Otherwise it remains closed (Figure 72).

Both the hydraulic press and the hydraulic jack make use of check valves which permit flow in but one direction. In Figure 72, two check valves open and close alternately as a result of pressure differences on their two sides alternately set up in the input cylinder as the input piston is moved up and down.

In modern hydraulic systems specially designed valves are used to control the direction of flow of liquids. With the development of modern hydraulic pumps, it was discovered that valves could be

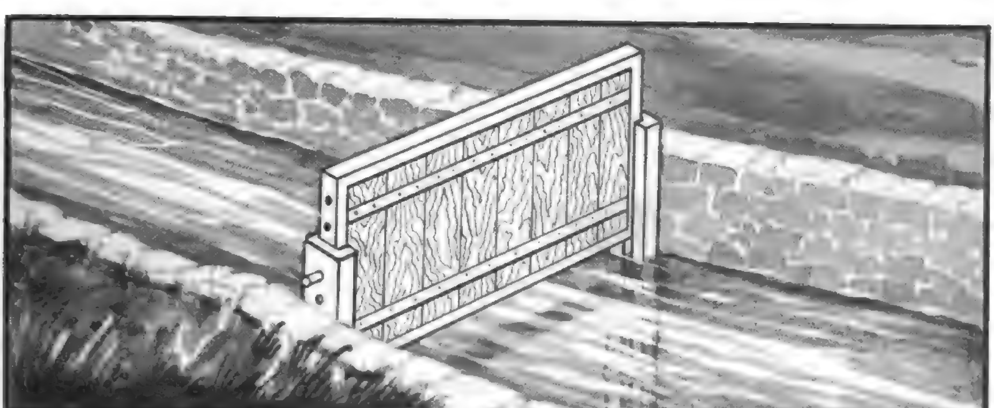


Figure 71

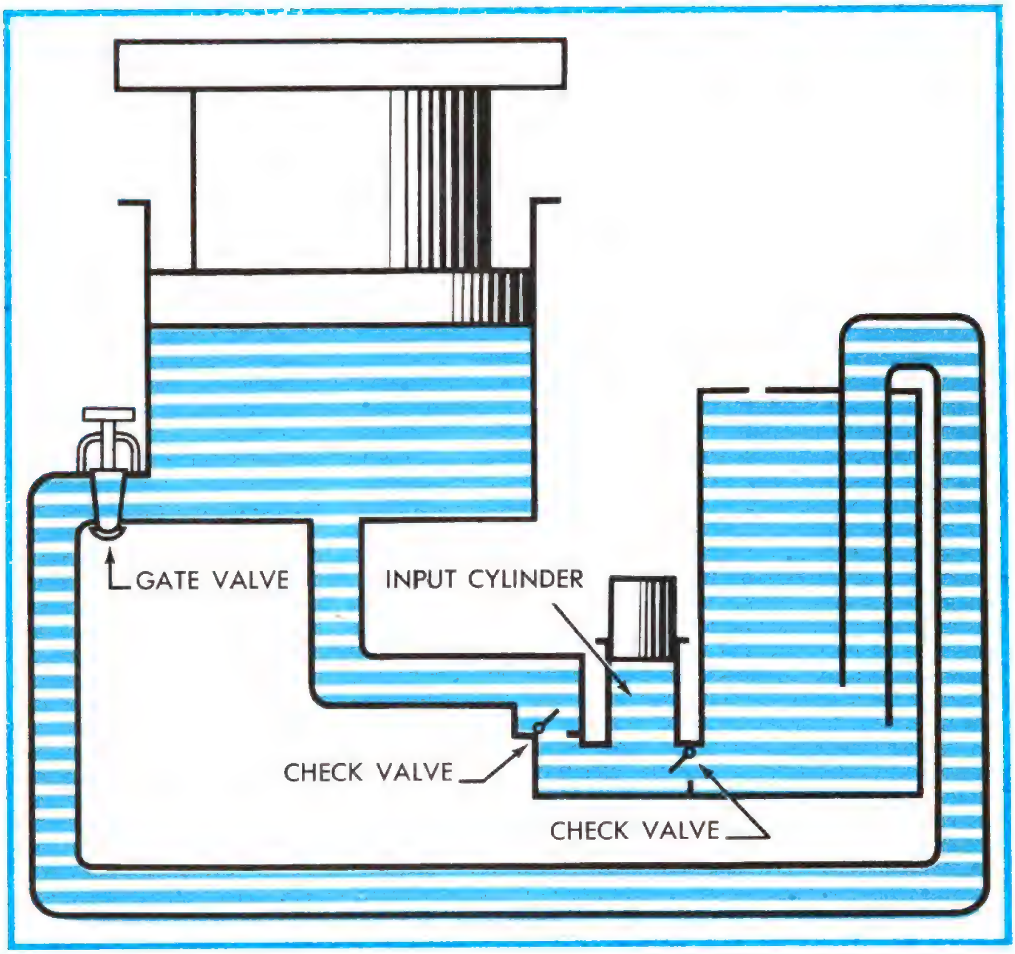


Figure 72

used to control system pressures. This led to the invention of relief valves to prevent a temporary dangerous rise in pressure, and pressure regulating valves to reduce the pressure in a main circuit for use in a branch circuit.

Classification of Valves

All valves can be classified as *simple, compound* or *directional* valves, according to their method of operation. A simple valve requires only *a single internal motion* for its operation. Liquid acting on one side of the valve, for example, opens it against the resistance of gravity or a spring; or the valve is raised manually by turning a screw so that a passage for the liquid is opened up. A compound valve involves *a combination of internal motions* for its operation.

Directional valves are used to control the direction of flow of liquids along two or more paths. This chapter describes simple valves. Compound valves are described in Chapter 6, while directional valves are described in Chapter 7.

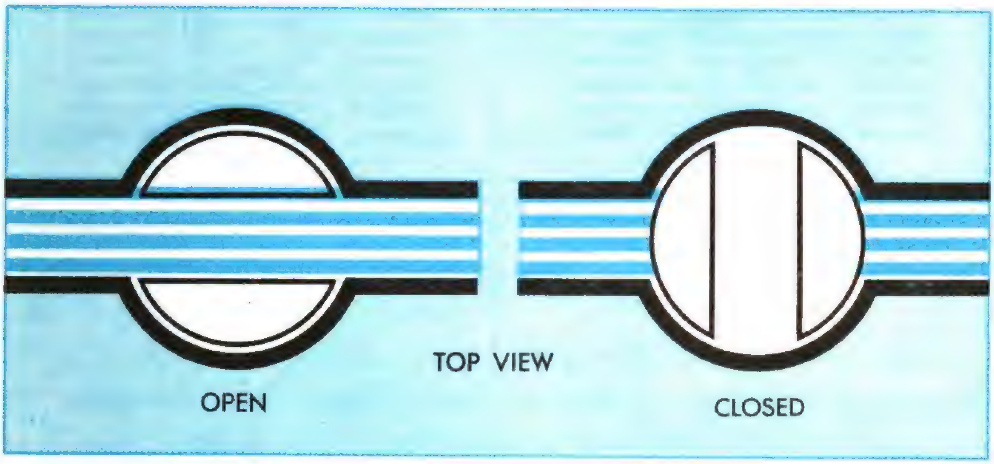


Figure 73

Simple valves can be grouped under the following heads:

a. *Cocks:* The valve plug is permanently seated in the pipe, and has an opening running through it to permit flow when the valve is open. The valve plug can be turned around in place so as to cut off flow (Figure 73).

b. *Gate valves:* A wedge-shaped gate stands outside the line of flow, but can be moved across the flow to cut it off (Figure 74).

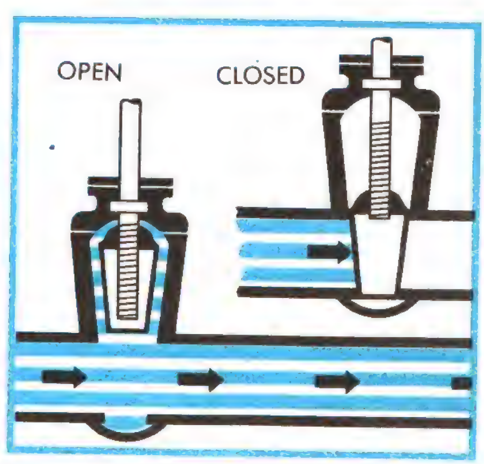


Figure 74

c. *Globe valves:* The movable controlling member is supported from its center, so that it can be raised or lowered across the flow (Figure 75). This type of valve necessitates a change in the direction of flow.

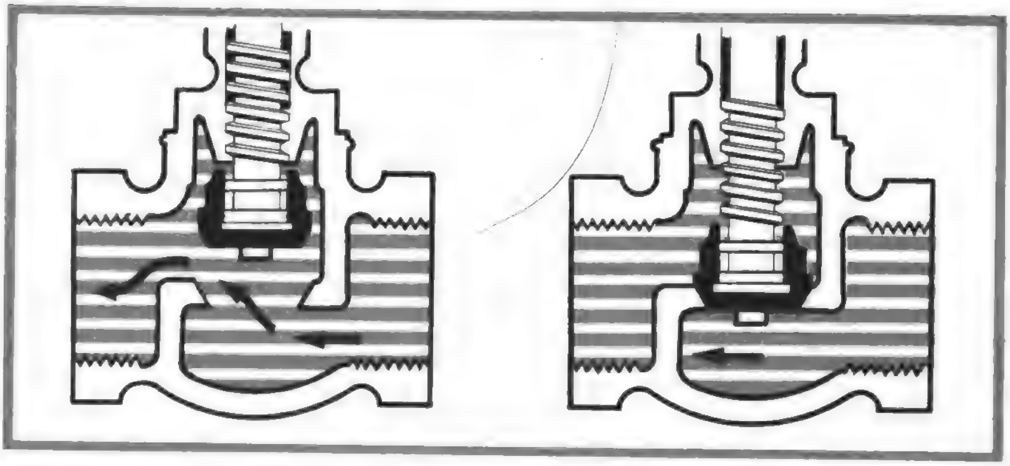


Figure 75

d. *Check valves:* Here the valve permits flow in one direction, and prevents flow in the opposite direction (Figure 76). It is opened and closed by differences of pressure on its two sides. Closure is sometimes assisted by a spring and/or by gravity.

e. *Relief valves, safety valves and automatic air vent valves* are really check valves used for special purposes. Their normal working position is closed, and they open only under special conditions.

Different Types of Simple Valves

Cocks. Figure 77 gives an exploded view of the elements of a cock. The bottom section, which stands in the pipe line, is cut in half. The top of the plug extends up through the gland, and can be turned with a wrench. In other types the plug terminates in a handle.

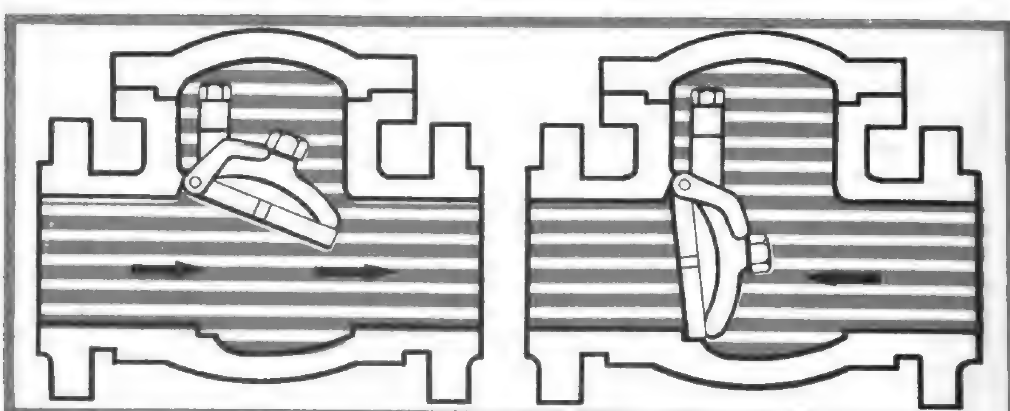


Figure 76

When the cock is open liquid flows through the hole in the plug. Flow is stopped by a quarter turn of the plug, which is marked to show whether the cock is open or closed.

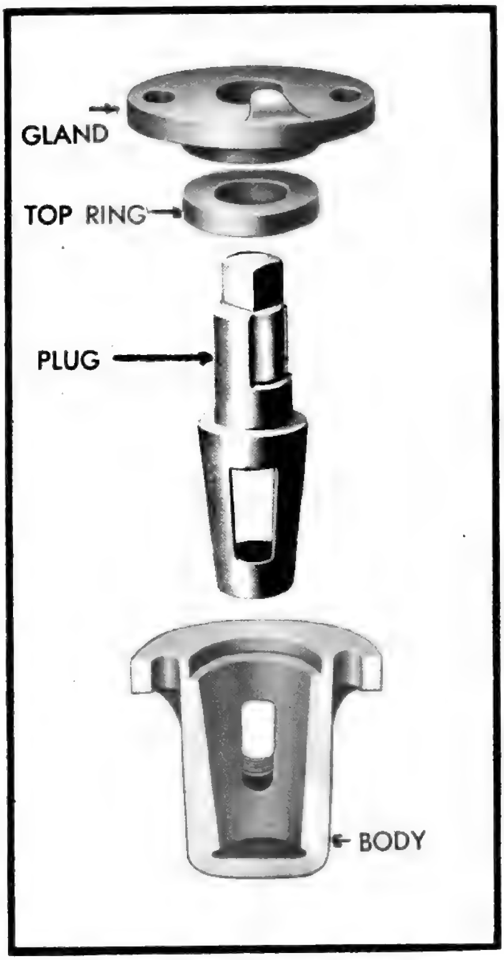


Figure 77

Although the inside surfaces of cocks are machined to give close contact, the meeting of metal with metal offers dangers of sticking due to seizure. When cocks stand normally in an open position the parts of the plug providing seal are not directly in contact with the liquid; but when the cock is normally closed the liquid will act on one of the sealing surfaces. Under ordinary conditions, however, cocks can easily be opened or closed.

Cocks are not designed to be used in a semi-open position. This is especially true if grooves in the wall of the cock body are filled with packing, as for example vulcanized asbestos, to separate metal from metal. In a partly opened position this packing would eventually wear away. In any event unequal wear is encouraged if the cock is left in a partly open position. Small cocks are sometimes used to free a system from accumulated air. The cock is opened so that the air can escape. When liquid begins to flow continuously the cock is closed.

Gate valves. In this type of valve, flow is controlled by means of a wedge or gate which can be moved up and down across the line of flow by a hand wheel, to open or close the passage. The principal elements of this type of valve are schematically indicated in cross section in Figure 78. *A* shows the pipe connection and the outside structure of the valve, while *B* shows the wedge or gate inside the

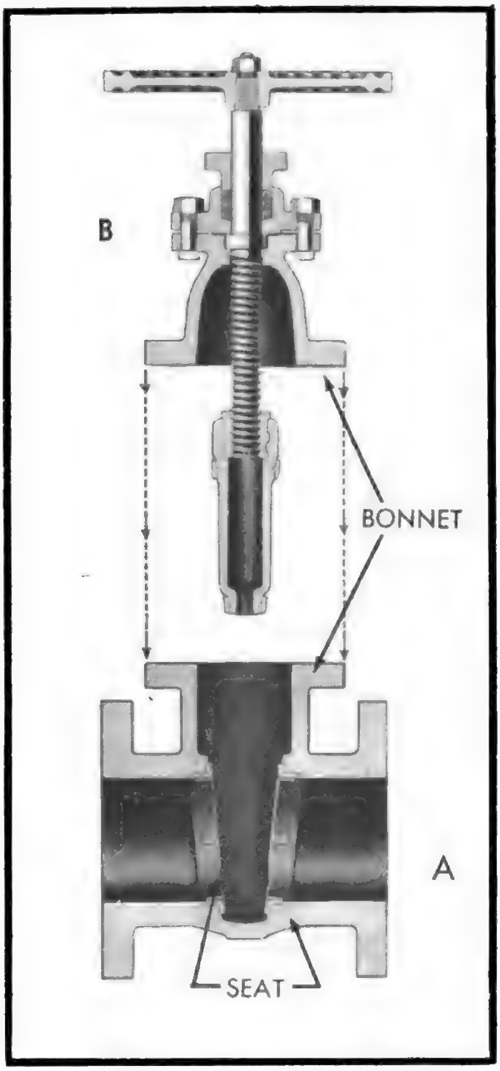


Figure 78

valve, and the stem to which the gate and the hand wheel are attached. When the valve is open, the gate stands up inside the bonnet, its bottom flush with the wall of the pipe. When the valve is closed, the gate blocks flow by standing straight across the pipe, where it rests firmly against two seats extending clear around the pipe.

Gate valves permit straight flow and offer little or no resistance to the liquid when the valve is completely open. Although they are often left partly open to throttle flow, they are intended for all-or-nothing-at-all use. If the valve is left partly open its face may be eroded, since it will stand in the line of flow, where liquid can act upon it. Gate valves are not easy to open or close if the pressure is high.

The gate of the valve can be a solid or a hollow wedge, or it can be made out of two facing disks. The wedge type is satisfactory for smaller valves under low pressure, although wedges are sometimes difficult to get tight and will leak when worn. By using disks a better closure can be provided, since the disks can be forced apart, snug against the valve seats, as they are moved into position. One arrangement for doing this is shown in Figure 79. One of the two facing disks composing the valve has been removed to show how the valve is constructed. Two cams with arms extending outward stand opposite each other on slanting surfaces in the space between the disks. As the disks move into position the arm of each cam engages a lug on the body of the valve and is turned on

the slanting cam bearing surface, forcing the disks against the valve gates during closure.

Gate valves are made with three types of stem connections. In Figure 80 the stem screws down into the valve gate as the valve is opened. In this type the stem does not rise or fall outside the body of the valve as the valve is opened or closed. In Figure 81, the stem rises outside the valve as the valve is opened, but the stem screw operates inside the body of the valve. In Figure 82, the stem screw operates at the level of the hand wheel, so that the stem rises independently of the wheel as the valve is opened. This is called the outside-screw-and-yoke type of valve.

Valves with rising stems are used when it is important to know by immediate inspection whether a valve is open or closed. This is the case, for example, in automatic sprinkler systems, where valves are

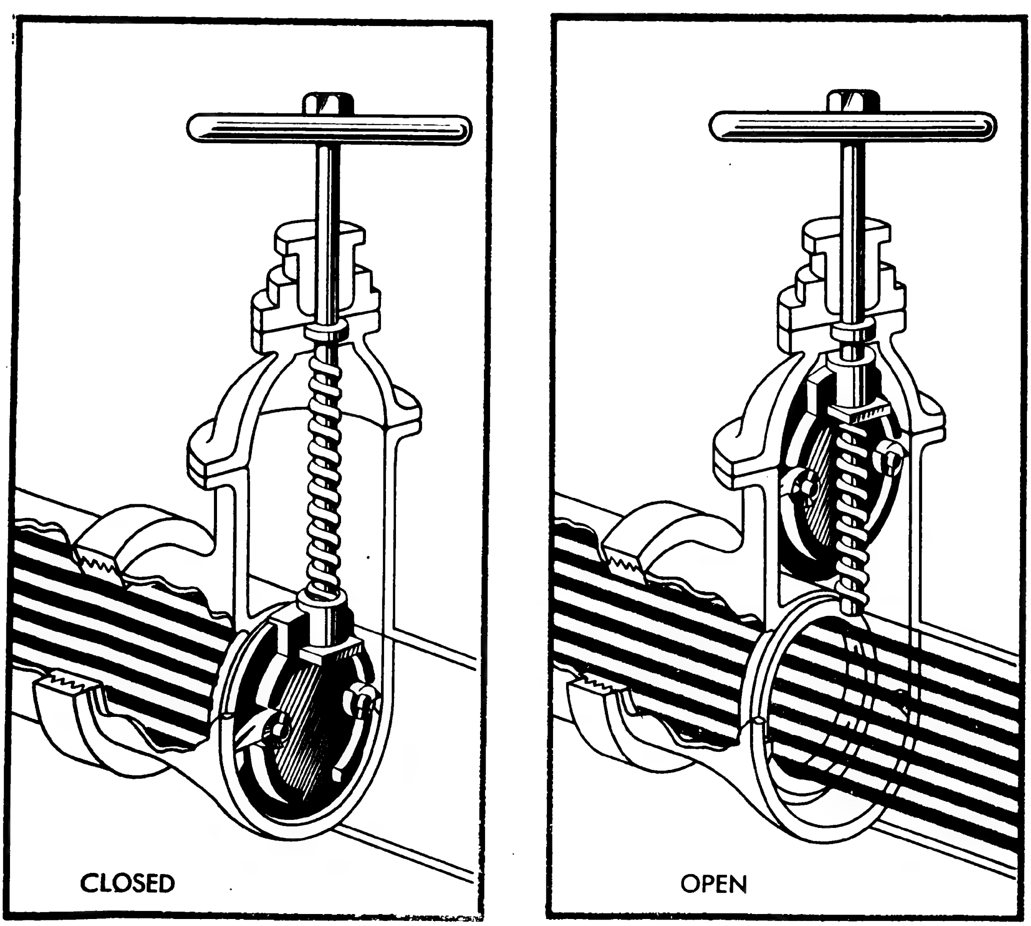


Figure 79

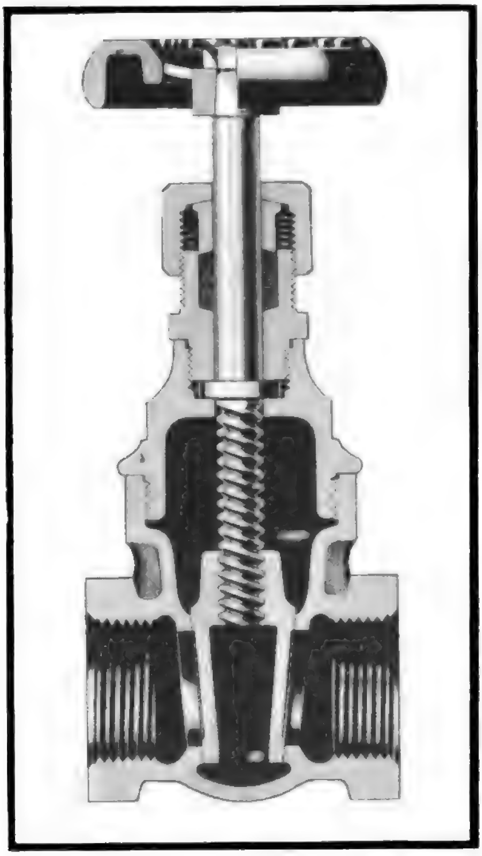


Figure 80

ordinarily kept open. The non-rising stem type is least likely to leak, and needs less room. Care must be taken not to force valves with non-rising stems. When it is desirable to remove the stem screw threads beyond the reach of corrosion, the outside-screw-and-yoke type can be used.

Globe valves. Here the controlling member of the valve, called the disk, is screwed directly on the end of the stem. The valve is closed by lowering the disk onto the valve seat. Since the liquid flows equally on all sides of the center of support when the valve is open, there is no unbalanced pressure on the disk to cause uneven wear and tear. An exploded diagram of a globe valve is given in Figure 83. Closure can be accomplished in a number of ways—as for example by a sharply tapered plug, a ball, a plug with a 45 degree seat, or a disk, made of metal throughout and regrindable while seated, or replaceable when worn, or fitted with a renewable composition seat held

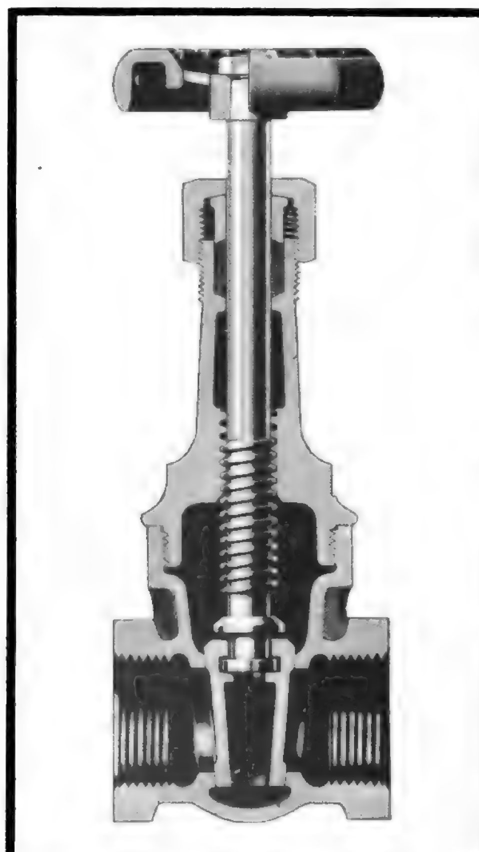


Figure 81

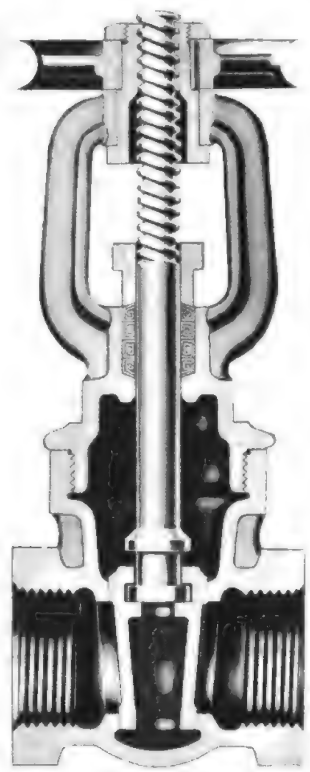


Figure 82

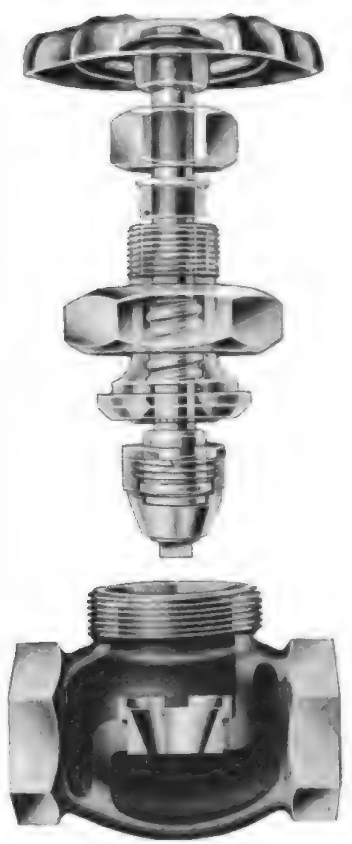


Figure 83

Figure 84

in place by a nut. The taper of the plug, the depth of the tapered hole, and the hardness of the material used at the seat will depend on the conditions of use. Figure 84 shows a variety of disks used in globe valves.

Needle valves end in a tapered point (Figure 85), so that the valve can be opened or closed very gradually. They are used to control flow into delicate gauges, which might be injured if liquids under great pressure were suddenly delivered to them; to control end operations of a

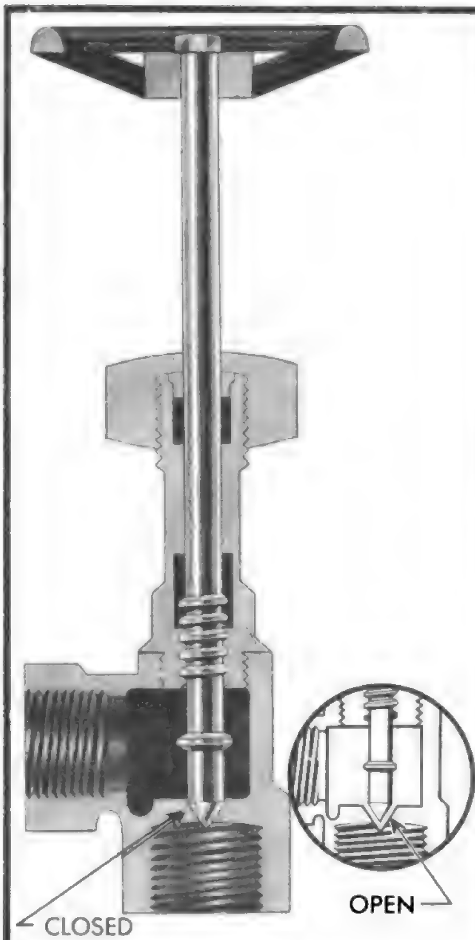


Figure 85

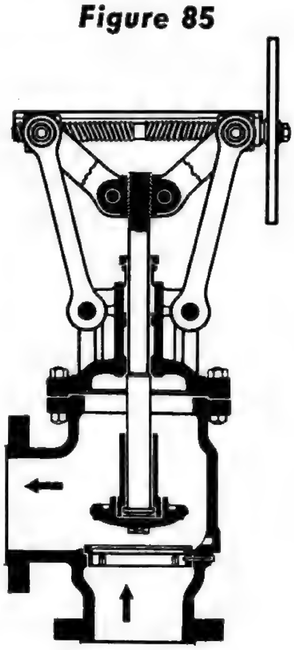


Figure 86

work cycle, where it is desirable that a work motion be brought slowly to a halt; and at other points where precise adjustments of flow are necessary and where a small rate of flow is desired.

Flow must change direction when passing through a globe valve, but the valve can safely be kept in a partly open position to control volume of flow, since pressure is evenly distributed around the center of support of the valve. The valve should always be installed with pressure against the face of the disk.

When the operating pressure of a globe valve, or in fact of any hand operated valve, is very high, a toggle operated gear may be necessary to permit closure (Figure 86). As the hand wheel is turned in a clockwise direction the leverage set up by the toggle linkage is increased, and it becomes possible to force the valve disk into contact with its seat.

Check valves. Valves of this kind permit flow in but one direction, since they close against reversed flow or when flow ceases. The force of the liquid in motion opens a check valve, while it is closed by back flow, by the action of a spring, or by

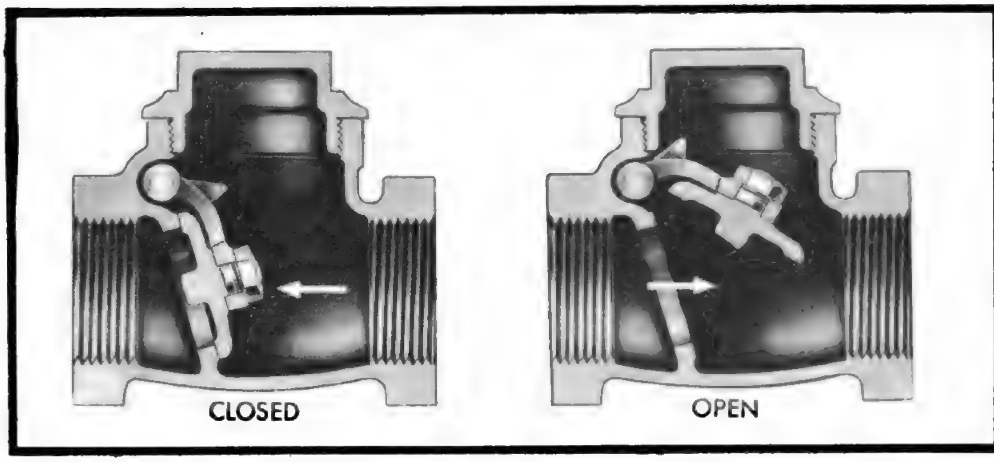


Figure 87

gravity. These valves should be installed so that gravity assists closure. Check valves are not manually operated.

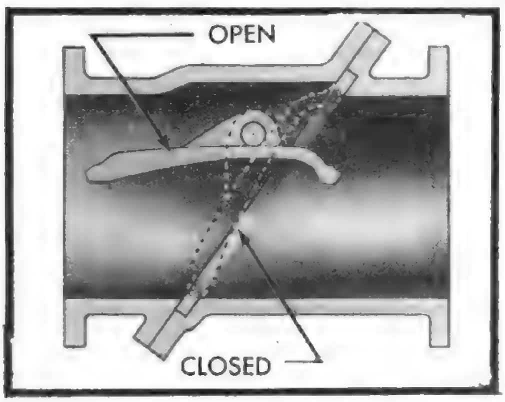


Figure 88

Figures 87-92 show a number of varieties of check valves: the ordinary swing type (87); a variation in which closure is accomplished by swinging the two ends of the disk in opposite directions against the seat (88); a vertical flow check valve (89); a hori-

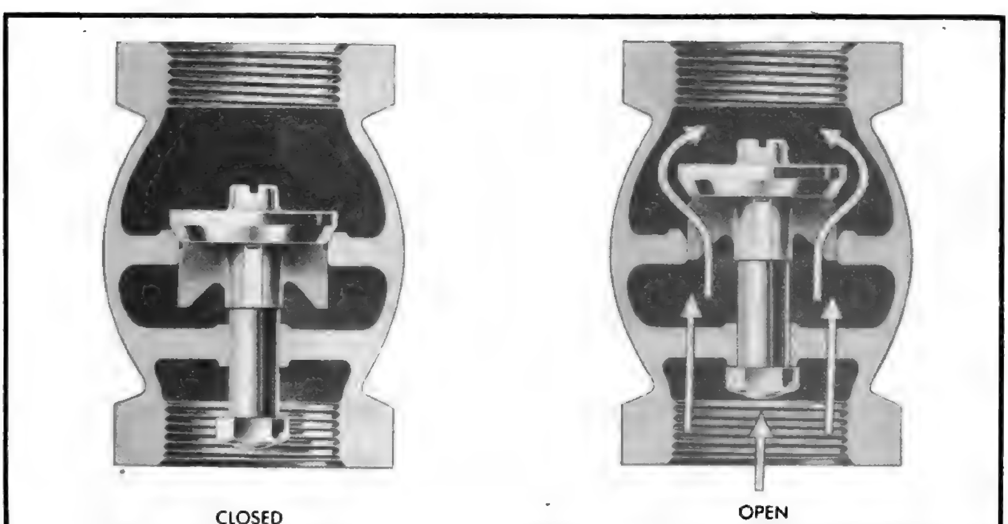


Figure 89

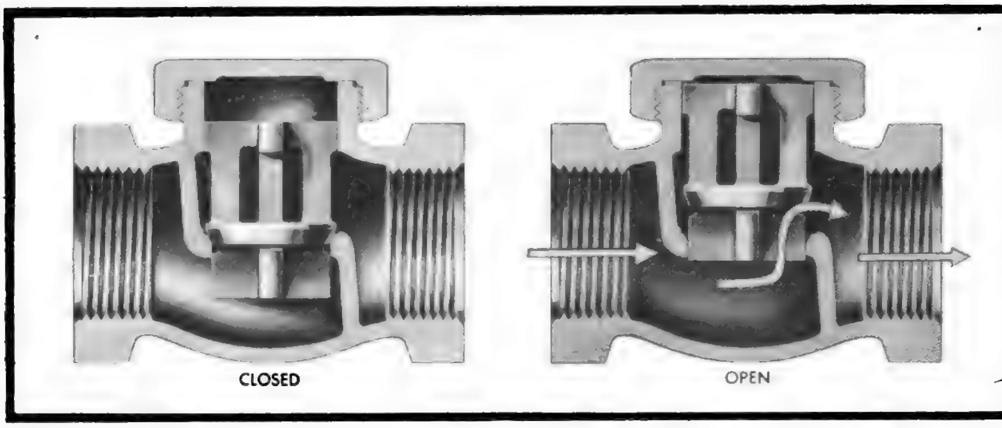


Figure 90

zontal flow valve requiring the flow to change direction (90); a piston type valve with a spring to hold the valve in closed position (91); and a ball type with spring (92).

The disk in many check valves is left free to rotate in place so as to distribute wear around its circumference. Swing type check valves offer less resistance to flow than lift types, but the lift type will close more tightly on low-pressure lines, especially if a spring is added. Conversely, however, it will open less easily. Lift valves are less likely to slam shut than swing valves. As pressure and velocity increase, however, lift valves are more liable to chatter and pound—a fault to which all kinds of check valves are liable.

Check valves are used in naval ordnance hydraulic systems for several purposes, the chief use being on replenishing lines. The valve is installed between a supply line and a main line which is subject to leakage. So long as the main line pressure is above that in the supply line, the check valve remains seated; but when pressure in the main line drops because of leakage, pressure from the supply line will open the valve and liquid will flow into the main line until pressure there is restored.

Relief and safety valves. It is often desirable to make certain that system pressure does not rise above a certain level. Relief valves are used for this purpose. Check valves like those shown in Figures 91 and 92 could be used to do this if they were equipped with a spring adjusted to compress when the pressure reaches a definite

level. Liquid is then by-passed out of the system or back into a reserve supply. The valve is controlled by means of an adjustment screw against which the top of the spring is seated (Figure 93). An inward adjustment of this screw increases the compression of the spring, making it able to resist a higher pressure. Relief

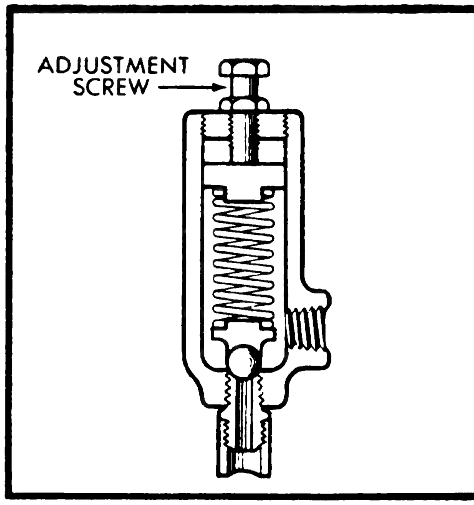
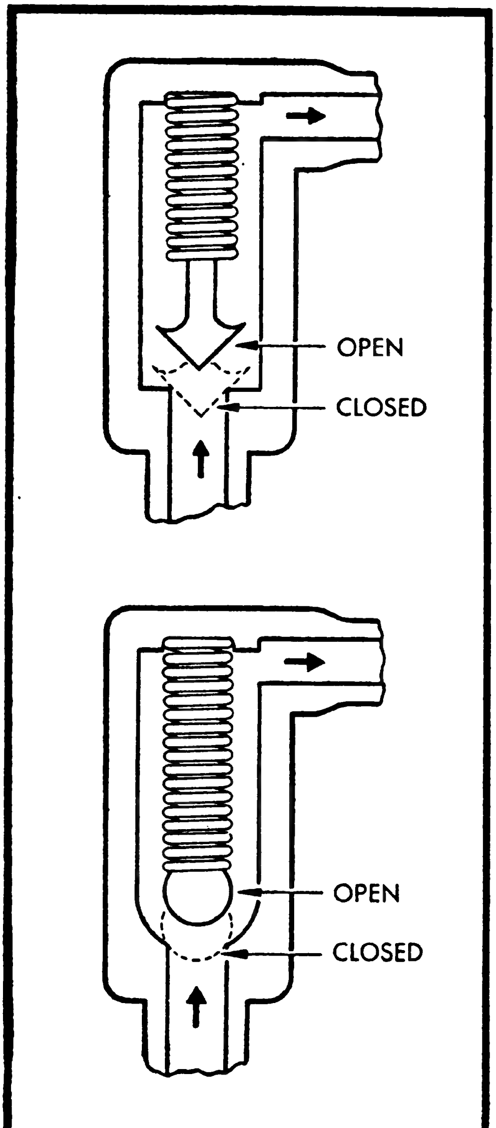


Figure 93

valves are usually installed near to the source of pressure, to protect as large a part of the system as possible.



Figures 91 and 92

Relief valves can be considered true safety valves only if they are large enough to handle the entire discharge of the mechanism in case of breakdown. Safety valves proper are designed to open only when the system is endangered, as for example if the operating mechanism stalls. They can take care of the entire load of the system, whereas relief valves are intended to relieve occasional excess pressures arising during the course of normal operations.

Various kinds of compound relief valves will be discussed in the following chapter.

Installation and Maintenace of Valves

The kind of metal to be used in a valve—brass, iron, steel, special alloys—will depend on the liquid employed and the working temperatures and pressures of the system.

Manufacturers of valves specify the range of safe usage for each of their valves, and these specifications should be carefully heeded.

Valves should be handled carefully. If a valve is dropped, a part vital to dependable operation may break or become deformed. Valves should not be located where they are exposed to damage from blows. They should be accessible, and it should be possible to open them fully. Valves should not be installed where they must carry the weight, sag or expansion of a line.

Valves should be in closed position during installation. Operating parts are less likely to become twisted, and areas of closure are protected. Overlength threading should be avoided on the joining pipe, so that the end of the pipe will not project into the valve and injure the valve seat. Sealing compound should be applied only to the pipe threads, and care taken that it does not get into the valve to injure the seat. The pipe should be cleaned before a valve is installed, and blown out again with compressed air after installation. In attaching a valve, the wrench should be used on the hex nut nearest the pipe to which the valve is being connected. The packing in a new valve should be tightened after it has been in use for a short time, but it should not be allowed to become tightly compressed.

Open and close valves *slowly*. Do not force the disk against the seat. If a valve refuses to seat properly, open it slightly so that sediment on the seat will be flushed away. If it still fails to seat, the system must be shut down and the valve disassembled to locate and correct the trouble. Valves that have been shut when hot should be tested for closure after the system has cooled.

Inspect all valves regularly. Keep them locked if there is any

chance of unauthorized manipulation. Do not turn a valve stem with a wrench. Do not remove the turning wheels from valves. Valves of all kinds should be tested at least four times a year.

In disassembling, be sure the valve is open. Do not use a long handled wrench to release the valve. A short wrench and a few light hammer taps over the threads are more effective, and will not twist the valve.

Use only the correct tools in reassembling, and again be sure the valve is in open position. Acquaint yourself thoroughly with the construction of the valve before putting it together. Repair parts should be ordered by giving the name of the manufacturer, the serial number on the valve, its size and type, and the name (and if possible the number) of the valve part.

The successful operation of a hydraulic system depends upon the proper operation of the valves in that system. The operator should know what valves must be open, and what valves closed. If the system has been operated previously the automatic valves, such as regulators, relief valves, safety valves, etc., should not be disturbed until found faulty or until time for periodic check-up. Always check up on the open or closed position of all hand operated valves before it is time to start the system.

Improper action of a hydraulic system can often be traced to faulty valve action. This is usually due to the presence of foreign matter in the valve seat, to the scoring or grooving of meeting parts, or to the plugging of openings. As a result the valve may stick, or may fail to close completely, so that the system registers low or fluctuating pressures, or even no pressure at all.

The usual remedy for such conditions, as in fact for practically all valve troubles, is to dismantle the valve, thoroughly clean all parts, replace those that are damaged, and reassemble. Any valve can easily be taken apart if care is taken to notice just how the different parts are related. Sometimes foreign matter can be blown out of a valve, especially a relief valve, by momentarily increasing the pressure in the system so that the particles will be forced out of the way.

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Leakage through a valve may be due to foreign matter, to scoring of the valve, or to distortion of the valve seat because the valve is not strong enough for the system. A scored valve can be reground, or a new seat provided. A distorted valve part will probably have to be replaced.

If the valve leaks through the stuffing box, the gland must be set up by tightening the gland nut, but not too much or the stem will stick. The stuffing box may need to be repacked. If the stem does stick, the adjustment screw over the stuffing box should be backed up to release the packing somewhat. The stem may stick because it is rusty or dirty, and therefore need cleaning.

The valve may have been closed too tightly when hot. It should open if the bonnet nuts—the nuts, that is, which join the upper and lower parts of the casing—are released slightly. If the valve has been opened too far when cold and then heated to a jam, it may be necessary to use a wrench to open it. The wrench should be applied carefully so that the valve seat will not be hurt. If the threads of the stem are stripped or burred, the stem should be replaced. Occasionally a valve functions badly because its spring has been broken.

QUESTIONS

1. Explain the principle of the gate valve.
2. What are the disadvantages of the gate valve?
3. In what ways does the globe valve differ from the gate valve?
4. What is the difference betewen simple and compound valves?
5. What advantages do needle valves offer over gate valves?
6. What is the purpose of a check valve?
7. What is the main use of check valves in naval ordnance?
8. What is the purpose of a relief valve?

9. What are some of the indications of relief valve trouble?

10. Name some of the causes of relief valve trouble.

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Chapter 6

COMPOUND AND PRESSURE REDUCING VALVES

This chapter describes compound and pressure reducing valves and considers how they operate, the uses to which they are put, and the casualties to which they are exposed.

How Compound Valves Operate

Simple valves differ from compound valves in the way they operate. A simple valve contains only one system of movable parts, while a compound valve contains two systems of movable parts whose joint action is responsible for the operation of the valve.

A simple relief valve, for example, consists of a check valve set by a spring to open at a certain pressure. As indicated in Figure 96, a compound relief valve consists of a pilot valve operating in exactly the same manner as a simple relief valve, and a main valve which opens and closes as a result of the action of the pilot valve. A full explanation of this action will be given later in this chapter.

The manner in which the most complicated valve opens and closes will always conform to the hydraulic principles outlined in Chapters 1 and 2. The valve will open or close because pressures are different over equal areas, because equal pressures are acting over unequal areas, or for both reasons together. In each instance inequality of opposed forces causes the valve to open or close. The forces may be set up by the opposition of hydraulic pressures, or by hy-

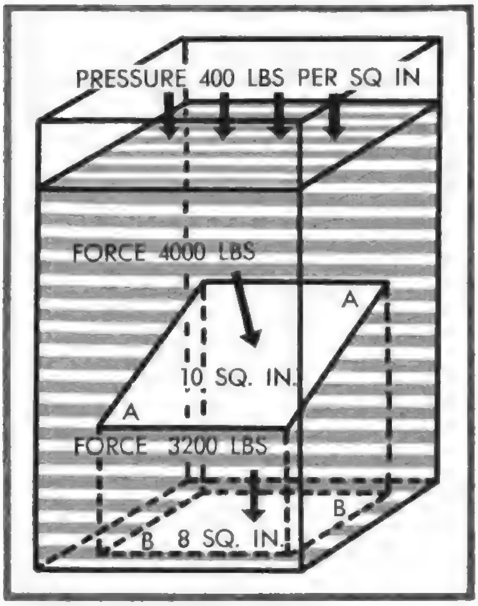


Figure 94

draulic pressure acting against a mechanical resistance, as for example the resistance set up by a spring.

Like all forces, the force exerted in a certain direction on the face of a valve is always exerted on an area standing exactly *at a right angle to that direction.* Thus in Figure 94, if the liquid is under a pressure of 400 pounds per square inch and the slanting surface *A* has an area of 10 square inches, the force acting *in a slanting direction,* at a right angle to that surface, will be 4000 pounds. The force acting *directly downward* on surface *A,* however, will depend on the *horizontal* area *B* which is directly beneath *A.* If *B* has an area of 8 square inches, the *downward* force acting on surface *A* will be 3200 pounds (400×8).

Compound Relief Valves

Advantages of compound relief valves. As we already know, even when a hydraulic system is operating normally it may develop temporary dangerous excess pressures, as for example when an unusually strong work resistance is encountered. Relief valves are used to control this excess pressure. Their purpose is not to maintain flow or pressure at a given amount, but to keep pressure from rising above a definite level when the system is temporarily overloaded.

It is because simple relief valves sometimes are unsatisfactory in operation that compound relief valves were developed. A simple relief valve with a suitable spring adjustment can be set so that it will open when the system pressure rises to say 500 pounds per square inch. When it does open, however, the volume of flow to be handled may be greater than the capacity of the valve, so that pressure in the system may rise as much as several hundred pounds above

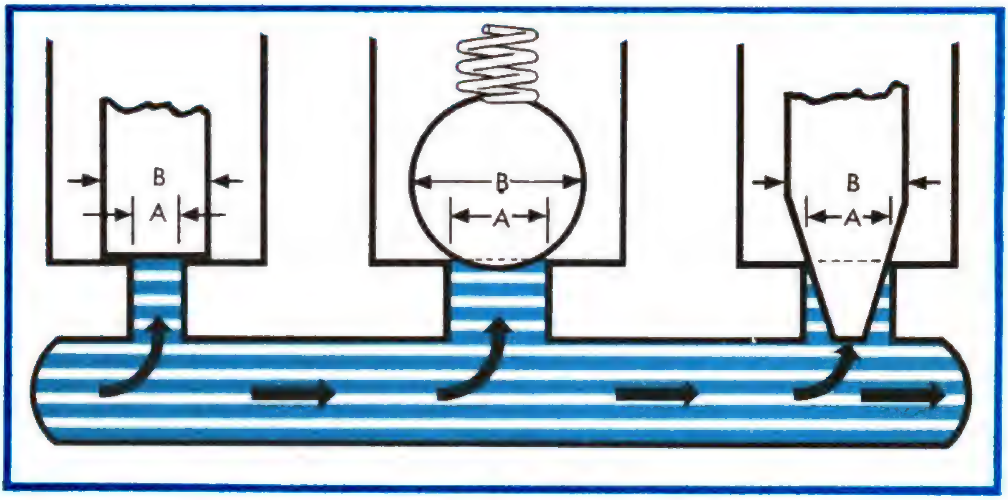


Figure 95

the set figure before the valve brings the pressure under control. A simple relief valve would be effective under these conditions only if it were very large. In that case, however, it would work stiffly and would chatter back and forth on its seat.

In addition, when the system pressure is reduced, the valve will close at a pressure *below* the set figure. As indicated in Figure 95, the seating surface of the valve must be wider than the pressure opening if the valve is to seat satisfactorily. The pressure in the system acts on the area of the valve open to it. In each case in Figure 95, the force exerted *directly upward* by this pressure when the valve is closed depends on the horizontal area across the valve at *A*. The moment the valve opens, however, the upward force exerted by the pressure will depend on the horizontal area of the valve at *B*, which is necessarily greater than the area at *A*. This leads to an upward jump in the action of the valve immediately after it opens, and it also sets up a greater force opposed to the closing of the valve than was needed to open it. As a result the valve will close at a lower pressure than the pressure at which it opened. We have here an example of the differential areas discussed in Chapter 1. The same pressure acting over different areas produces forces proportional to the areas.

For example, the valve may be set to open at 500 pounds per square inch. If the upward area at *A* exposed to this pressure when the

valve is closed is 0.5 square inch, an upward force of 250 pounds (500×0.5) will be exerted on the valve at the moment of opening. With the valve open, however, the pressure acts on the area at *B*, which might be 1 square inch. The upward force would then be 500 pounds, or double the force at which the valve actually opened. For the valve to close, pressure in the system would have to fall well below the figure at which the valve opened. The exact pressure at which the valve would close would depend upon certain geometrical relations of the particular shapes involved which cannot be further discussed here.

It follows that simple relief valves are liable to open and close rapidly as they *hunt* above and below the set pressure, causing pressure pulsations and undesirable vibrations in the system, and producing a noisy chatter. Compound relief valves use the principle of action of simple relief valves for one phase of their action—that of the pilot valve—but provision is made to limit the amount of liquid that the pilot valve must handle, and thereby avoid the weaknesses of simple relief valves.

Structure and location. Figure 96 distinguishes the pilot valve, main valve, and valve body in two compound relief valves—the Vickers hydrocone valve and the Northern simplex valve. The inlet of the valve is connected to the pressure line of a hydraulic system, between the pump and the point of work, while the outlet of the valve

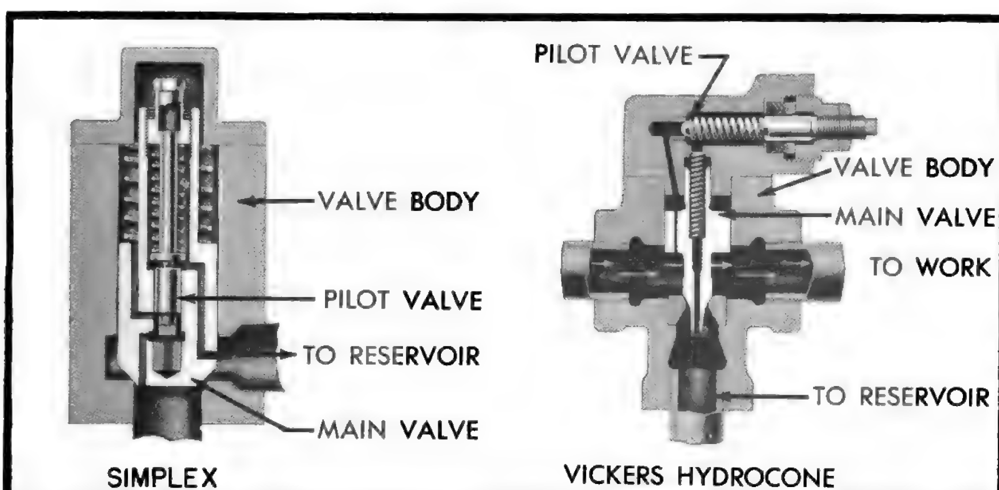


Figure 96

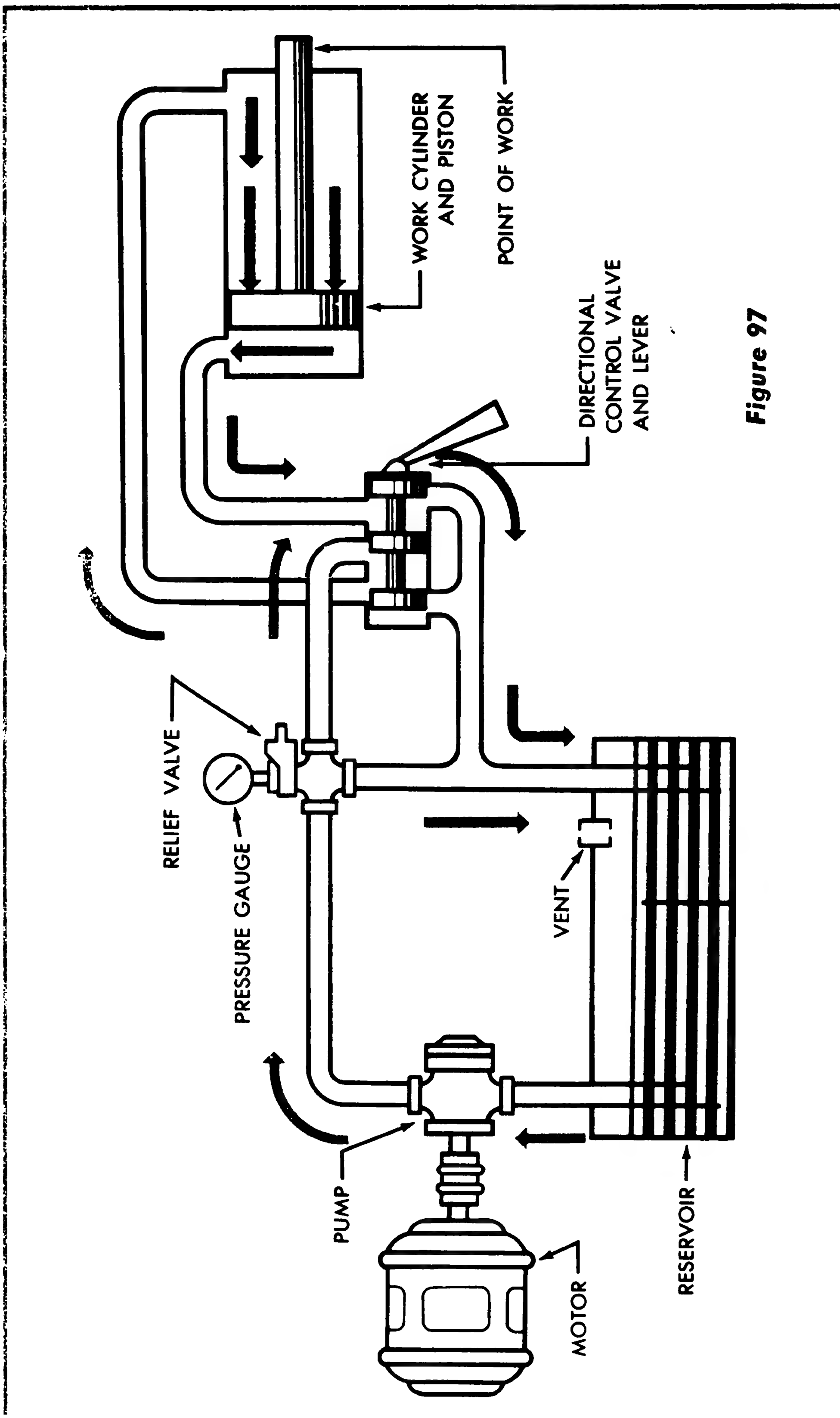

WORK CYLINDER AND PISTON
POINT OF WORK
DIRECTIONAL CONTROL VALVE AND LEVER
RELIEF VALVE
PRESSURE GAUGE
VENT
PUMP
MOTOR
RESERVOIR

Figure 97

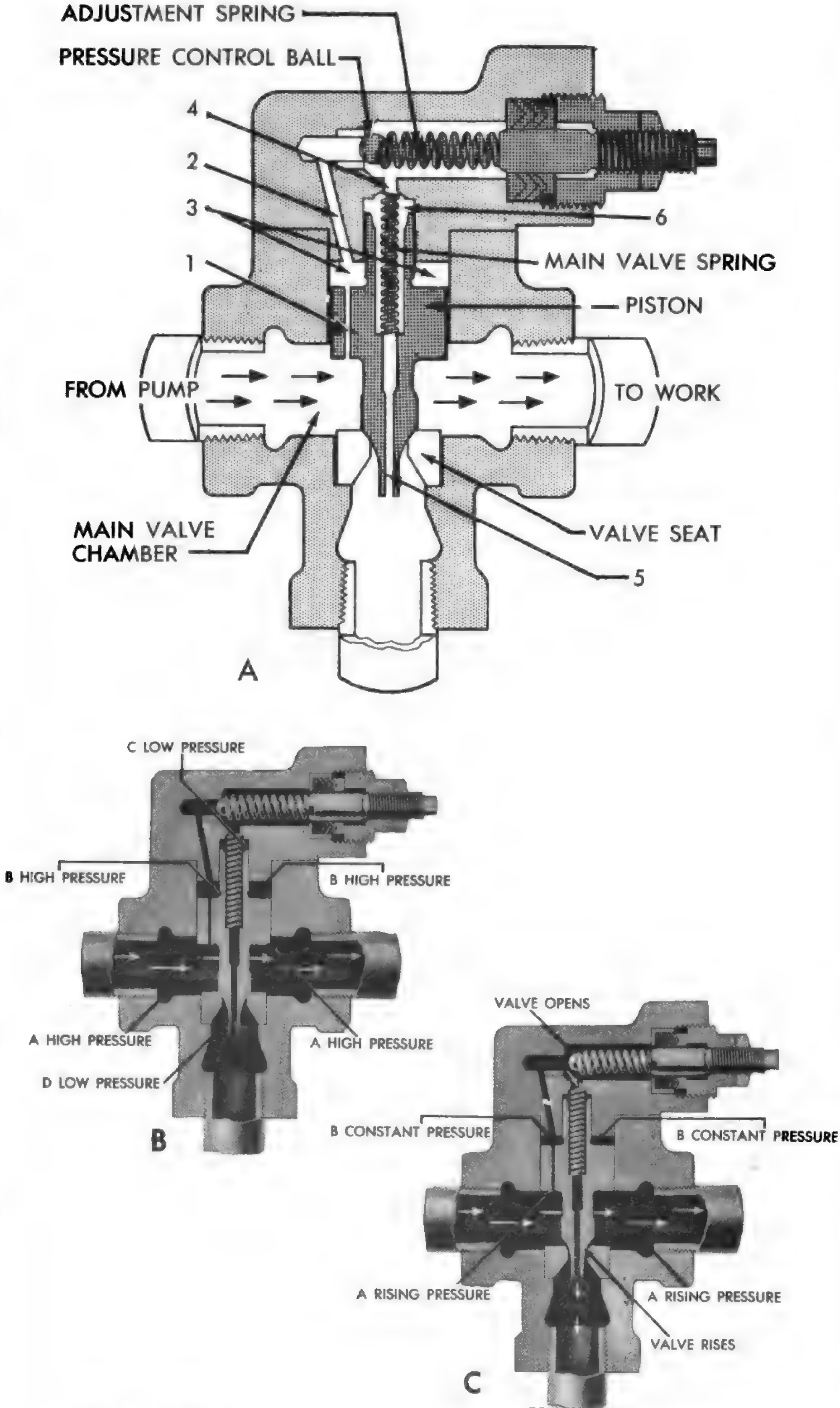
is connected to the reservoir of the system. A general indication of a typical set-up is given in Figure 97. A pump driven by a motor receives liquid from a reservoir and sends it along a pressure line past the compound relief valve and on to the point of work. The liquid is returned to the reservoir from the work area through another pipe line.

Under normal conditions liquid flows past the valve and no valve action takes place. If pressure in the system rises above the setting of the pilot valve, however, the pilot valve is forced open, and this leads to the opening of the main valve, in a manner shortly to be explained. A portion of the liquid in the pressure line is thus diverted to the reservoir, thereby relieving the excess pressure in the system. These valves are always installed on the high-pressure side of the pump, and as close to it as convenient.

Vickers hydrocone relief valve: structure. Figure 98A shows the relations between the different parts of this valve. It consists of a main valve and a pilot valve, the latter set by an adjustment spring to open at a desired pressure. The two valves are connected with each other, with the main pipe line and with a drain line to the reservoir by a number of passages and chambers.

The main valve extends down through the main line of flow of the high-pressure liquid. Flow past this part of the valve is always possible, since it fills only a portion of the pipe. The lower end of the main valve terminates in a cone, the upper part of which registers in a conical shaped seat to prohibit flow to the reservoir when the main valve is closed. Above the line of flow the main valve expands to form a piston. Above this piston is a chamber, which is connected by a small passage to the main line and by a larger passage to the pilot valve. The areas on the upper and lower surfaces of the piston are approximately equal. The main valve, finally, is equipped with a light spring.

With both valves closed, the entire assembly is divided into two pressure systems, as shown in Figure 98B. Pressure on one side of the pilot valve is normally high, since it is determined by pressure in the main line. Pressure on the other side of the pilot valve is



SITUATION AS MAIN VALVE OPENS

Figure 98

normally low, since it is determined by pressure in the reservoir line.

The pilot valve is connected on its high-pressure side to the main pipe line by passages *1* and *2*, between which stands chamber *3* (Figure 98A). The upper face of the piston of the main valve forms the bottom surface of chamber *3*, and is always subject to the pressure contained in that chamber. The size of passage *1* is restricted so that no more liquid can pass through it than the pilot valve can handle without being overloaded.

The pilot valve is connected on its low-pressure side with the pipe line to the reservoir by passages *4* and *5*, down through the center of the main valve. Between these passages stands chamber *6*. The main valve has an upward extending stem, the top of which forms the bottom of chamber *6* and is therefore subject to the pressure therein. All of the passages on the low-pressure side of the pilot valve are larger than passage *1*, so that any liquid that gets through passage *1* when the pilot valve is open can pass through all the other passages and be drained off to the reservoir without building up pressure on the low-pressure side of the assembly.

Operation. The main valve will remain closed so long as a greater force acts downward on it than acts upward. The relation of forces when the valve is closed is shown in Figure 98B. The opposed areas *A* and *B* are approximately equal, and both are under the same high pressure. The opposed areas *C* and *D* are equal, and both are under the same low pressure. Hydraulically, therefore, the main valve is in balance. There remains, however, the unbalanced downward force of the light spring, which keeps the main valve seated so long as the pilot valve remains closed.

As long as there is no flow through passage *1*, the pressure in chamber *3* will remain equal to that in the main pipe line. If pressure in the main line rises so that pressure against the control ball of the pilot valve becomes greater than the resistance of its adjustment spring, the pilot valve will open and permit liquid to flow through passages and chambers *1, 3, 2, 4, 6,* and *5* back to the reservoir. The ball valve will act as a simple relief valve releasing liquid from

chamber *3* to the reservoir and, within the practical limitations of a simple relief valve, will hold the pressure in chamber *3* down to the limit determined by its adjustment spring.

Pressure relations when pressure in the main line rises above the set of the pilot valve will be as shown in Figure 98C. A pressure difference will be set up on the opposed areas *A* and *B* of the main valve. Area *B* will be held at a constant pressure by virtue of the fact that the pilot valve is open, whereas the pressure on area *A* will tend to increase. The moment the resultant upward hydraulic force becomes great enough to overcome the small downward force of the main valve spring, however, the main valve will rise from its seat and liquid will be discharged directly from the main valve chamber to the reservoir. This will continue until enough liquid has been drained off to bring down the pressure in the main valve chamber to the setting of the pilot valve adjustment spring. At this time pressure above and below the main valve is equalized, and the main spring causes the valve to seat.

It will be seen that this valve overcomes the greatest limitation of a simple relief valve by limiting flow through the simple valve to such quantities as it can satisfactorily handle. This limits the pressure above the main valve, and enables the main line pressure to open the main valve. In this way the system is relieved when an overload condition exists.

Venting the hydrocone valve. The main valve opens because the force acting upward on area *A* is greater than the force acting downward on area *B* (Figure 98C). If a means can be devised to reduce the pressure in chamber *3* and passage *2* considerably below the pressure in the main line between the pump and the valve, whenever we wish to do so, the hydrocone valve can be used to secure low working

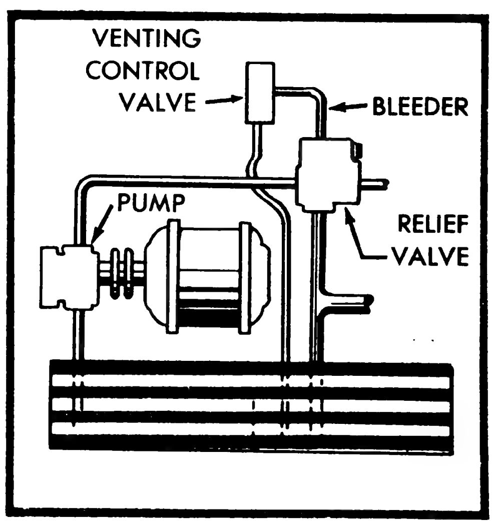


Figure 99

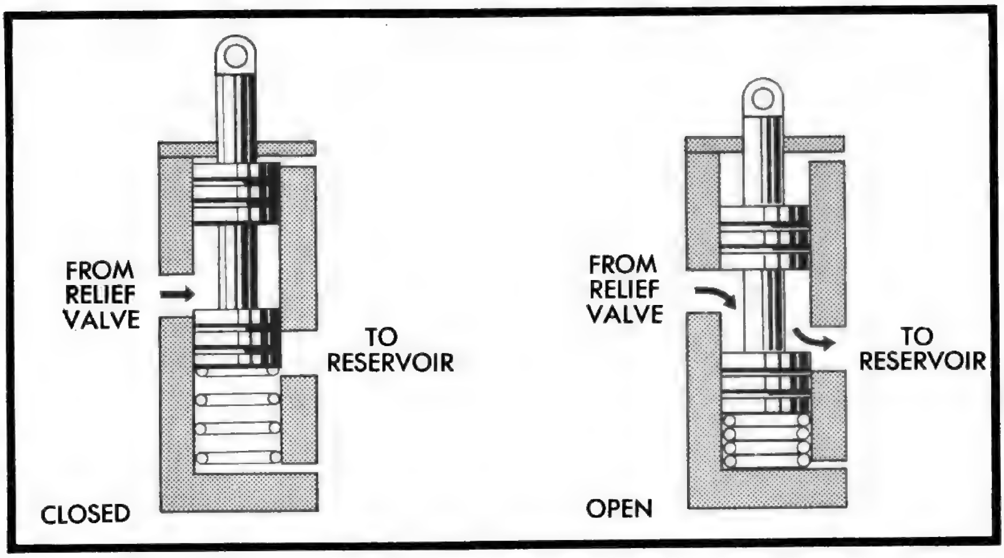


Figure 100

pressures in a system also delivering high pressures, without changing the speed of operation of the pump. Then the greater part of a work stroke can be driven under high pressure and the stroke completed under low pressure, thereby cushioning the end of the stroke; or the system can be relieved between work strokes.

This can be done by connecting chamber *3* and passage *2* of Figure 98 to atmospheric pressure—that is, to the reservoir—by way of a restricted passage called a bleeder, as shown in Figure 99. When the venting control valve is closed, the hydrocone valve acts as a compound relief valve. When the control valve is open, liquid pressure from the main line carried up into chamber *3* will cause a small volume of liquid to flow continuously from chamber *3* through the bleeder back to the reservoir. Because of the unequal forces acting on opposed faces of the main valve, at *A* and at *B* in Figure 98C, it will be unseated, and liquid will continuously discharge past the main valve to the reservoir, venting the system pressure down to the desired level. In this manner a hydrocone valve set to open at pressures up to several thousand pounds per square inch can be vented down to atmospheric pressure or very little more, as may be required.

To open the venting control valve and start the venting operation, its piston is lowered by hand, by a cam, or electrically. Figure 100

shows a venting control valve in the closed and in the open position. When the control valve is closed, its piston blocks flow through the bleeder. When the control valve is open, liquid flows through, from the upper chamber of the relief valve to the reservoir, to produce the venting action.

Vickers valves having model numbers ending with "V" do not allow venting to lower pressures than 60 pounds per square inch, while valves not so marked can be vented down practically to atmospheric pressure. The pressure at which the valve will vent depends upon the size of the bleeder and the strength of the main valve spring.

Northern simplex relief valve. This valve is manufactured by the Northern Pump Company, and is often found on their hydraulic

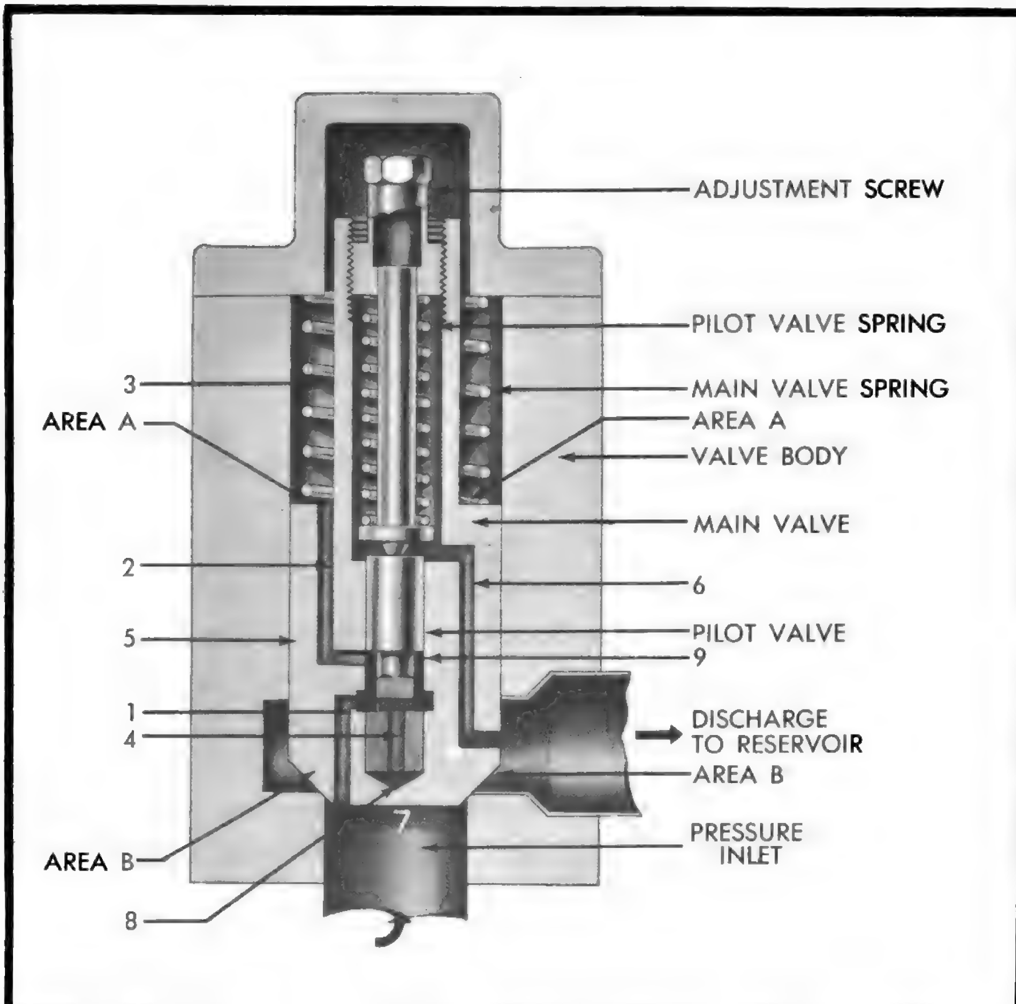


Figure 101

equipment. It works on the same principle as the Vickers hydrocone valve, in that the opening of a pilot valve enables hydraulic pressure in the system to set up an unbalanced force on the seating face of the main valve, thereby causing it to open. This result is brought about, however, in an entirely different manner.

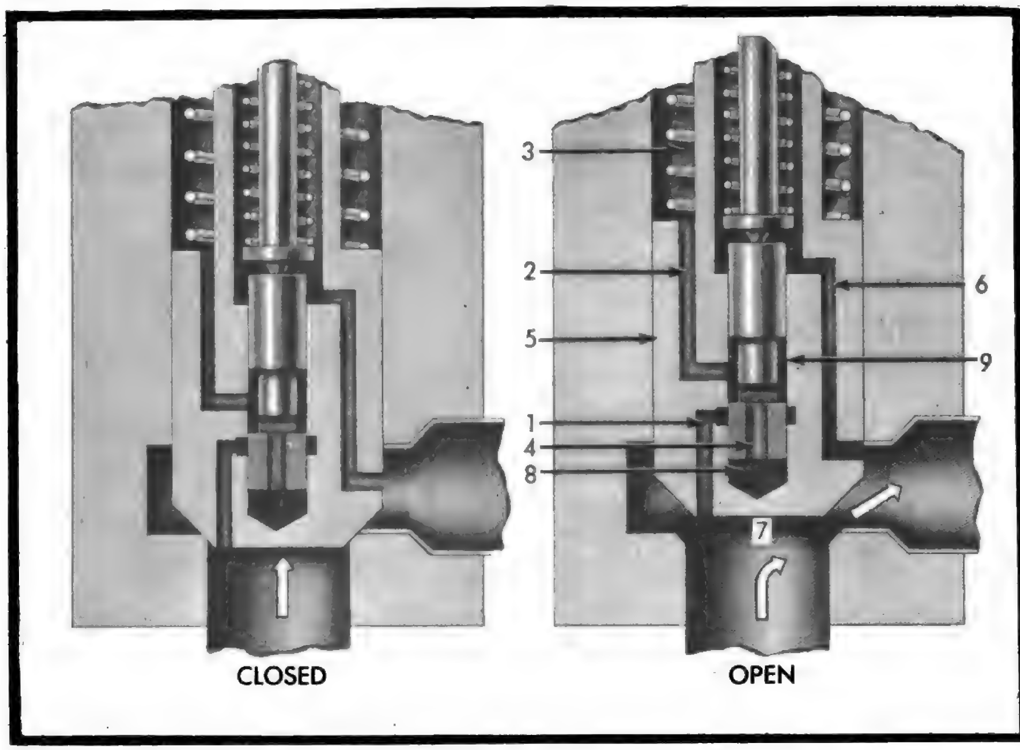
Structure. Figure 101 shows details of this valve in closed position. The principal working parts of the valve—the body, the main valve, and the pilot valve—are also shown in Figure 96. The pilot valve lies inside the main valve, and the two units are connected by passages which are basic to the action of the valve. Also important is the fact that a small amount of leakage exists between the main valve and the valve body.

In the drawing, the main valve and the lower part of the pilot valve have been cut away in order to show these passages. It is important to realize that the rectangles on either side of passage *4* are a physical part of the pilot valve. In fact they constitute the lower part of its piston.

With both the pilot valve and the main valve seated, passage *1* admits liquid from the main pressure line by way of passage *2* to region *3* over the main valve, including the region at the very top of the valve over the adjustment screw of the pilot valve. Passage *1* also admits liquid by way of passage *4* to region *8*, under the base of the pilot valve, as likewise to region *9*, under the upper part of the piston of the pilot valve. By this arrangement pressure in the main line is conveyed to the region *over* the main valve and *under* the pilot valve when both valves are closed.

In addition to the passages already noted, passage *6* connects the region over the piston of the pilot valve to the valve discharge, and therefore to the reservoir of the system. In this manner liquid which has leaked past the piston is prevented from hydraulically blocking the upward movement of the pilot valve.

Operation. With the system under normal working pressures, the main valve is held closed because the horizontal area *A* of the main valve on which pressure acts downward is twice the horizontal area



Figures 102 and 103

B on which pressure acts upward, so that the downward force on the main valve is twice the upward force. The only purpose of the main valve spring is to hold the main valve closed when the system is not under pressure.

Figures 102 and 103 show two stages in the action of this valve. Figure 102 shows the pilot valve open while the main valve remains closed, while Figure 103 shows the situation with the main valve also open. Let us suppose that the adjustment spring of the pilot valve has been set to compress when pressure in the system reaches 500 pounds per square inch. At this pressure the pilot valve will rise, and therefore will at first restrict, and shortly thereafter cut off the connection of passage *1* with the other open areas inside the valve. Regions *3*, *8* and *9* remain in communication with each other, however, and therefore are subjected to the same pressure of 500 pounds per square inch. In the succeeding action of the valve, pressure in these regions will remain substantially constant at 500 pounds per square inch.

As pressure in the main line increases above 500 psi, pressure in

region *8* will also increase, and the pilot valve will rise. This will further restrict the opening from passage *1*, and leakage past *5* down the body of the main valve will cause pressure in region *8* to decrease. Thus the pilot valve will descend, opening up passage *1*, pressure in region *8* will again increase, and the above described action will once more take place. In this manner the pilot valve regulates the pressure above the main valve, keeping it approximately constant and equal to the spring setting of 500 psi.

When the pilot valve lifted, the pressure on the base of the main valve, at *7* in Figure 101, was also 500 pounds per square inch. But the force acting upward there was only half the force acting downward on the main valve, because of the differences in areas. If pressure in the main line rises above 1000 pounds per square inch, however, the force acting upward on the main valve will become greater than the force acting downward, with the result indicated in Figure 103, where the main valve has opened so that liquid will drain from the main line through the discharge to the reservoir. This will only be possible, however, because of leakage from region *3* down through the very small space *5* separating the main valve from the valve body. Without this leakage the main valve would be hydraulically blocked. The volume of liquid held in region *3* also acts as a cushion to the action of the valve, to a degree preventing chatter.

Here as with the Vickers hydrocone valve, more satisfactory control of pressure is obtained by using a simple valve to hold a constant pressure on one face of the main valve, while a variable system pressure is being produced on the opposite face. The simple valve is protected from overload by limiting the flow it must handle, the rest of the relief flow being transferred to the main valve. Because of the different construction of the two valves, however, the pilot valve spring of the Vickers valve must be set to respond at the desired relief pressure, while with the simplex valve it must be set to respond at approximately half the desired pressure. This is a difference of more importance to the manufacturer than to the user, since the latter will always adjust the valve in terms of its response to main line pressure, without considering the pressure at which the pilot valve spring will compress.

The range of application of simple relief valves and of the Northern simplex valve overlaps, since the lowest pressure at which the simplex valve can be set is around 100 pounds per square inch, while the simple relief valve is adequate for pressures up to about 250 pounds per square inch. Northern simplex valves have been used at pressures as high as 4000 pounds per square inch. When the Northern simplex valve is used within its designed capacity, the pressure in the system will never exceed the relief setting by more than about 5 per cent, no matter what the volume of flow.

Joint use of two relief valves. In some hydraulic systems—to be described later—a hydraulic pump-motor combination performs a work operation in such a way that each of the two pipe lines joining the hydraulic pump to the hydraulic motor must in one phase carry liquid under high pressure from the pump to the motor, while the other is carrying liquid under low pressure from the motor back to the pump. Since each line may be under pressure, each must be protected from overload. This can be done by installing two oppositely facing compound relief valves, each set up to relieve high pressure from one of the lines. Each valve discharges the excess liquid into the other line, as shown in Figure 104.

Northern duplex relief valve. A single duplex valve has been devised to perform the functions of the two valves in Figure 104. This

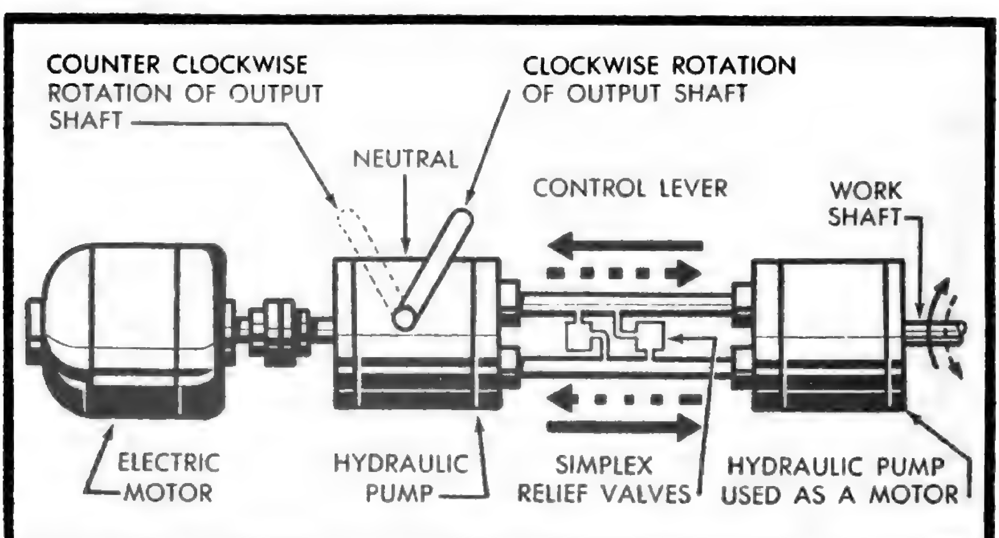


Figure 104

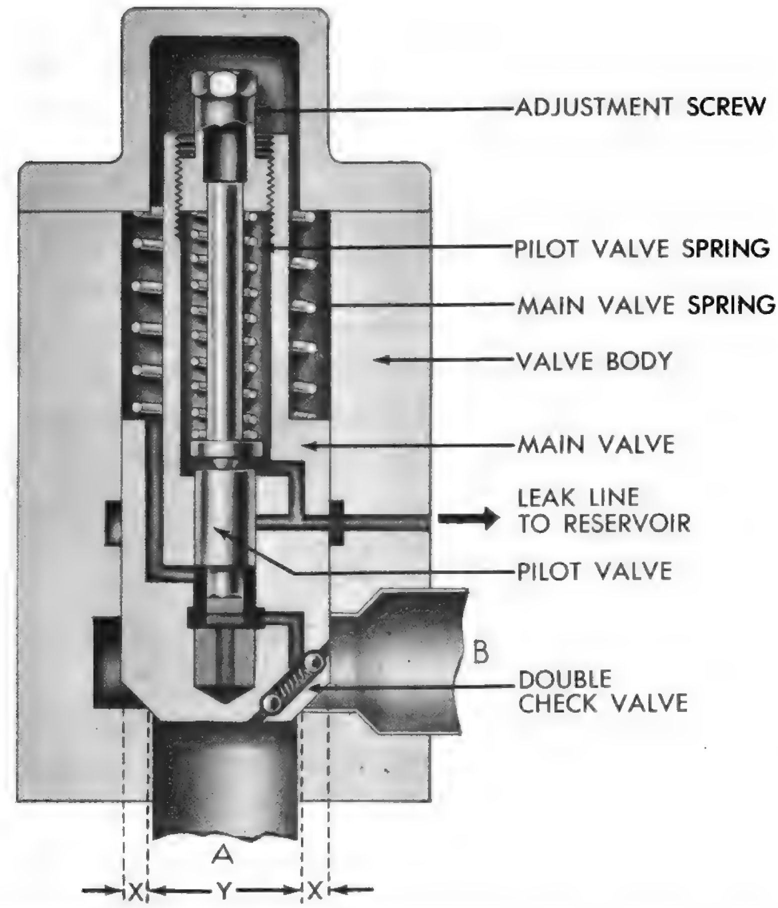


Figure 105

valve rather than the simplex is used in Navy installations. The duplex valve is essentially similar to the simplex valve, but there are two differences. In the first place, a double check valve at the base permits liquid to flow to the pilot valve from either *A* or *B*, whichever is under the greater pressure (Figure 105). The pilot valve and the main valve operate under ordinary conditions of excess pressure as they do in the simplex valve, but liquid is admitted by way of either passage *A* or passage *B*, and is discharged by way of the other passage when the main valve opens.

There is a second point of difference between the ordinary simplex valve and the duplex valve used by the Navy. As a further measure for the relief of excess pressures; a bleeder passage connected

directly with the reservoir has been added (Figure 105). This passage remains closed even after the pilot valve has risen sufficiently to cut off flow from the main pressure line into the areas above the main valve and under the pilot valve. If an extreme overload is placed on the system, such as a complete stall, the main valve will not rise quickly enough to limit main line pressure to the desired setting, and the pressure will therefore continue to rise. The main valve will open to some degree, however. As a result of the main line pressure increase, pressure in the chamber over the main valve will be increased above the value at which the pilot valve spring is set. This increase in pressure will be communicated to the chamber under the pilot valve, since this chamber and the region over the main valve are connected with each other. This will force the pilot valve to rise still further. The action will be assisted by leakage past the main valve back to the main line.

When the pilot valve has risen sufficiently to open its side passage to the bleeder line, the oil over the main valve will quickly be released through the bleeder to the reservoir. The sudden drop in pressure on the upper side of the main valve will allow the main valve to rise sufficiently to relieve the extreme excess pressure in the main line. In this manner the bleeder line positively prevents excessive pressures from building up in the system in the event of an extreme overload.

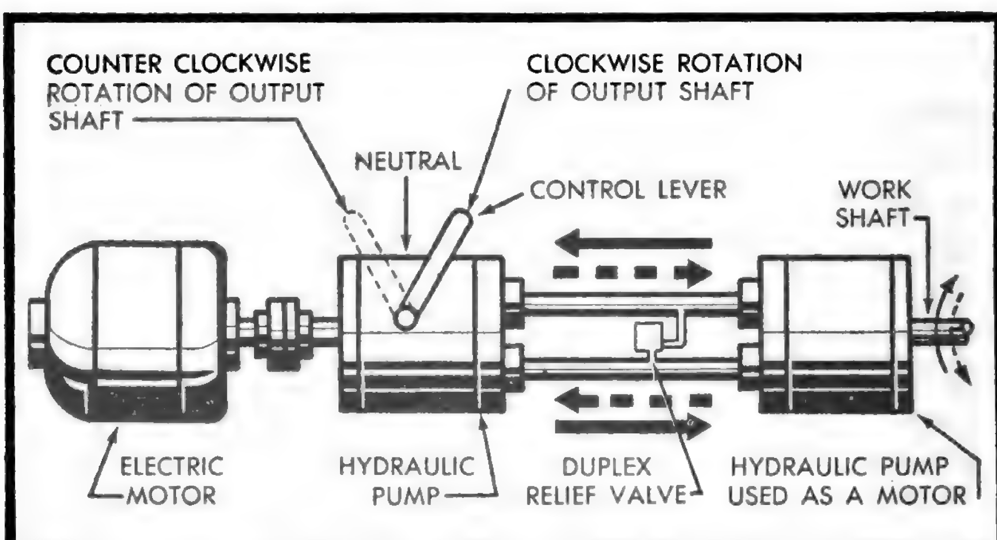


Figure 106

The areas of the main piston at x and at y (Figure 105) must be so arranged that the upward force exerted at a given pressure will be the same, no matter whether A or B is the pressure line. Otherwise the main valve would open at a different pressure for flow in each direction. The excess pressure bleeder line and the leak line from the upper part of the pilot piston must be directly connected to the reservoir in the duplex valve, since neither A or B is directly connected to the reservoir.

A single duplex valve can be used instead of two simplex valves in the installation diagrammed in Figure 104. It would be placed between the two pipes and connected as shown in Figure 106. It would then be able to relieve flow from either line into the other according to the direction of flow from the pump to the hydraulic motor.

Casualties with relief valves. An improperly operating relief valve will overheat, and will operate sluggishly, erratically, or at the wrong pressure. The most frequent cause of trouble is when foreign matter lodges in openings or on seats. The valve will usually clean itself if the pump is started, and the adjustment screw on the pilot valve is backed off a little by turning it counter-clockwise so that the pressure control spring responds to a lower pressure. The adjustment screw should never be completely removed with the system under pressure. After the flow of liquid has cleaned the valve, the adjustment screw should be carefully reset by means of a pressure gauge. The relief valve should be set to open at about 25 per cent above the maximum normal operating pressure.

If this procedure fails to clean the valve, the power unit driving the pump should be stopped, the valve cover removed, and the operating piston and spring taken out so that the inside of the valve can be cleaned. The piston should move freely up and down in its cylinder. In reassembling the valve, each part should be replaced in proper position, without forcing, and the cover sealed with a gasket in good condition. The cover screws should be tightened uniformly.

In checking relief valves, look for scored or worn parts. A valve

spring may be broken. The pressure control ball of the Vickers hydrocone valve can be reached by opening the pilot valve, removing the washer, oil seal and gasket, and taking out the spring and ball with a magnetized rod. In disassembly it is important to notice the order in which the different parts have been removed. If the pressure control ball must be replaced, a standard 5/16-inch ball bearing can be used. The ball can be reseated with the help of a brass rod by tapping it gently with a light hammer.

Pressure Reducing Valves

How they work. Pressure reducing valves are needed when the main line delivers high or fluctuating pressures, although a lower constant pressure is required on a branch line. System pressure acts upon a spring, the spring compresses, and the piston moves across the line of flow, blocking pressure down to a figure determined by the adjustment of the spring. Although some pressure reducing valves are really simple valves, how they work can best be grasped after an understanding has been gained of compound relief valves.

Logan pressure reducing valve. This valve is made in two models—8035, which permits only one direction of flow; and 8036, where the direction of flow can be reversed. They are used only in oil systems and operate under a maximum pressure of 1500 pounds per square inch. The 1/2-inch valve can handle 10 gallons per minute, the 3/4-inch 16 gallons, the 1-inch 26 gallons, and the 1 1/4-inch 50 gallons.

A schematic layout of both models is given in Figure 107. In model 8035 the body consists of a specially shaped solid piston supported between a heavier conical spring at its top and a lighter spring at its base. The lower spring merely holds the piston in place. Liquid enters the valve as indicated, from the high-pressure line, flows down underneath and around the narrow part of the piston, and out the other side of the valve into the low-pressure line. The piston is acted upon by the upper conical spring under which it stands, by liquid pressure, and to a small degree by the lower spring. The higher pressure on the inlet side acts upward on the lower surface of the piston at *A*, and downward on the slanting surface of the valve at

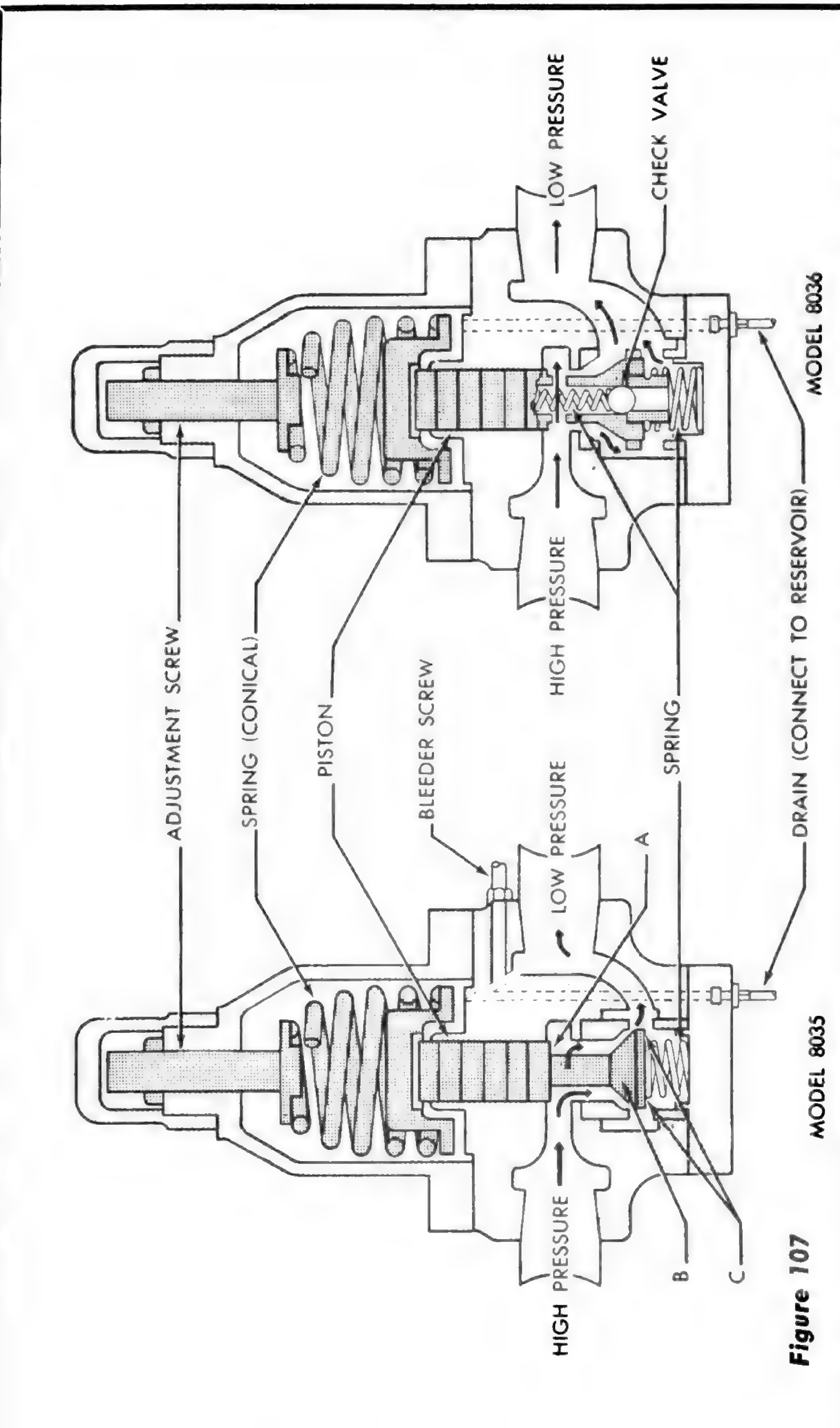

ADJUSTMENT SCREW
SPRING (CONICAL)
PISTON
BLEEDER SCREW
HIGH PRESSURE
LOW PRESSURE
CHECK VALVE
SPRING
DRAIN (CONNECT TO RESERVOIR)
A
B
C
MODEL 8035
MODEL 8036

Figure 107

B. Since the upward and downward components of the forces acting on these areas are equal, forces on the inlet side of the valve are in balance. This means that the action of the valve is independent of the pressure on the inlet side. Pressure there can vary widely without affecting the operation of the valve.

Pressure against the bottom of the piston at *C* acts against the resistance offered by the upper conical spring. This spring is set to compress at the pressure desired on the low-pressure side of the valve. If an increase in pressure on the high-pressure side sends more liquid into the area under the piston, increasing the pressure there, the piston will be forced upward, restricting flow from the high-pressure line so that the pressure on the low-pressure side of the valve will be returned to the desired level. If the valve momentarily closes too far, because of the increase in pressure under the piston, the increased throttling of flow produced when the passage for liquid is still further restricted will act to restore the pressure for which the valve was set.

If flow through the valve were stopped or nearly stopped, the valve would practically have to close to maintain the proper pressure. Under these conditions the slightest leakage from the valve would make it impossible to control the pressure properly. This is prevented by the bleeder, which can be adjusted by a screw so that it bleeds off greater or smaller amounts of liquid from the low-pressure side of the valve and thus maintains a certain minimum flow greater than the valve leakage. When the bleeder screw is properly adjusted, the valve will not completely shut, but will throttle down the flow to the desired pressure. The bleeder must be adjusted in terms of the main line pressure, the volume of flow, and the viscosity of the liquid.

In Model 8036 the piston is partly hollow at its base, and contains a ball check valve held inside the narrow part of the piston by a screwed-in seat. Small openings admit liquid to the area inside the piston over the check valve. Normal operation for the valve as shown, is for flow to pass from the left side to the right. As long as the pressure differential produces flow in this direction, the ball check

valve will remain closed and the entire valve will operate after the manner of Model 8035. If the pressure on the inlet side of the valve decreases below the pressure on the outlet side, however, the check valve will open, since the force exerted upward on the under surface of the check ball will be greater than the downward force on its upper surface. Liquid will then reverse its flow through the valve until pressure in the pipes on either side of the valve is practically equalized, when the check valve will be closed by its spring. Although flow in both directions is possible with this model, the pressure can be regulated in but one direction—that not requiring flow through the check valve.

Both models must be connected so that the high-pressure side of the valve is on the high-pressure side of the system. Both models are supplied with a drain to dispose of liquid that has leaked past the piston into the upper part of the valve, where the conical spring is located. The drain must be directly connected to the reservoir. It must not be allowed to clog, or high back pressure over the piston may blow out a gasket or do more serious damage.

The bleeder line in Model 8035 also must not become clogged. When beginning to operate the system, the bleeder adjustment should be completely opened for a short time so that the bleeder can clean itself out. The oil used should be of proper viscosity. Bleeding off will be hindered if the liquid is cold and thick. Both models can be dismantled for cleaning, or to replace worn or broken parts.

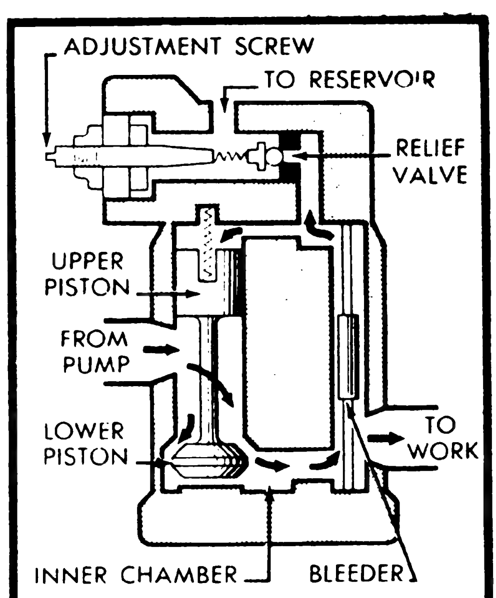


Figure 108

Vickers pressure reducing valves. A schematized diagram of this valve is shown in Figure 108. Liquid enters the valve as shown, moves past the lower piston into the inner chamber, and passes out of the valve. It also rises

past the bleeder into the upper part of the valve, where it comes into contact with the upper surface of the main piston and with the relief valve.

Since the opposed faces of the upper and lower pistons are very nearly equal, the forces acting on them are easily kept in balance by the spring, irrespective of pressures in the high-pressure line. Similarly, a balance is maintained on the low-pressure side between pressures on the upper side of the upper piston and on the lower side of the lower piston. The balance of low pressures is completely independent of the balance of high pressures.

Low pressure will depend on the degree of restriction to flow offered by the lower piston. But the exact position of the lower piston is determined by the adjustment of the relief valve. If pressure on the discharge side of the valve—pressure, that is, under the lower piston, in the inner chamber, up through the bleeder, against the upper side of the upper piston and against the relief valve—builds up to a point where the relief valve opens, some of the liquid in the upper part of the valve will be drained off. Since the flow of liquid to the upper part of the valve is restricted by the bleeder, a very small loss of liquid will cause a large pressure drop in this region. This is due to the fact that friction losses are large when liquid must flow through the bleeder. The pressure drop unbalances the forces acting on the upper and lower pistons, causing the piston to rise and restricting the flow still further until the pressure on the discharge side of the valve is reduced to the pressure at which the relief valve will close.

Here again we have a simple relief valve through which a limited flow is permitted, controlling a main valve by setting up an unbalanced hydraulic force in such a way as to produce a motion which restores the balance of forces. The net result is to hold the pressure on the discharge side constant at the setting of the relief valve. In this valve the relief valve will maintain the desired minimum flow so no other provision for flow is necessary.

This valve should be mounted vertically. It may need cleaning, either by backing off the adjustment screw of the pilot valve while

the pump is running, as previously described, or by dismantling the valve with the system out of operation. The relief ball may become worn, in which case a 5/16-inch ball bearing may be used to replace it; or a spring may occasionally need replacing. In replacing the relief ball, it should be reseated by tapping it lightly with a hammer by means of a rod placed against it.

QUESTIONS

1. What is meant by a compound valve?

2. Why is a pilot valve used in compound relief valves?

3. Why can compound relief valves be used for higher pressures than simple relief valves?

4. Why does the main piston of the Vickers relief valve and the Northern simplex valve rise from its seat?

5. What is meant by venting a valve?

6. How do pressure reducing valves differ from relief valves?

7. How does the check valve in the Logan pressure reducing valve 8036 permit reversal of flow?

8. Why must liquid be admitted to the upper surface of the Vickers pressure reducing valve?

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Chapter 7

DIRECTIONAL VALVES

In this chapter directional valves of the spool and rotary types are described, after which flow control valves and valve blocks and panels are briefly considered. The chapter concludes with a discussion of different methods of operating valves.

What They Are

Directional valves are designed for the specific purpose of directing the flow of liquids in hydraulic systems. It may be desired, for example, to perform a work operation by driving a piston back and forth in its cylinder. A directional valve whose movable parts change position so as to alternately introduce and drain off liquid from each end of the piston cylinder would make this possible.

Directional valves may operate hydraulically by differences of force set up on opposite sides of their movable parts, or they may be positioned by hand, by mechanical means, or by electric power. Sometimes two or more methods of operating the same valve will be utilized in different phases of its action.

Four-way valves are directional valves with 4 pipe connections: one from the pressure source, a second from or to the reservoir, and the other two connected to the work cylinder or water. This is the type of valve most commonly used.

Directional valves are of two designs—spool and rotary. In the spool design a specially shaped sliding piston opens and closes passages through the valve as it is moved back and forth in its cylinder. In the rotary design a round block or core is rotated inside a sleeve. Recesses and passages in the block make the desired connections between passages in the sleeve, which lead to different parts of the system. Valves of each of these designs will now be discussed in more detail.

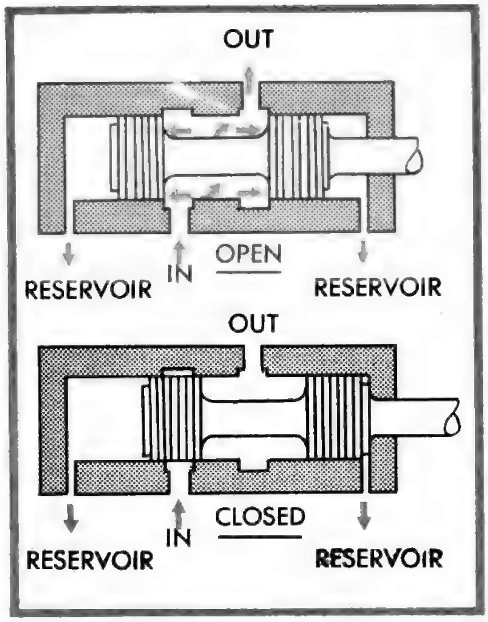


Figure 109

Spool Directional Valves

Two way spool valves. A two-way sliding valve is shown in open and closed positions in Figure 109. As the piston is moved back and forth, it either allows liquid to pass through the valve or prevents flow. A typical use of this valve was shown in Figure 99, where it was used as a venting control valve with the Vickers hydrocone valve.

The piston of the valve illustrated in Figure 109 cannot move back and forth by differences in hydraulic pressure set up within its cylinder, since the forces there are in balance. As indicated by the arrows against the piston heads, the same pressure acts on equal areas on their inside faces; and when the input passage is blocked the piston blocking it is acted on all around its circumference by the same pressure.

A number of features common to most spool type valves can be noted in Figure 109. The drain openings at either end of the cylinder are needed so that back pressure will not be built up in the cylinder to hinder movement of the piston. When spool valves become worn they may lose balance because of greater leakage on one side of a spool than on the other. In that event the piston would tend to stick in

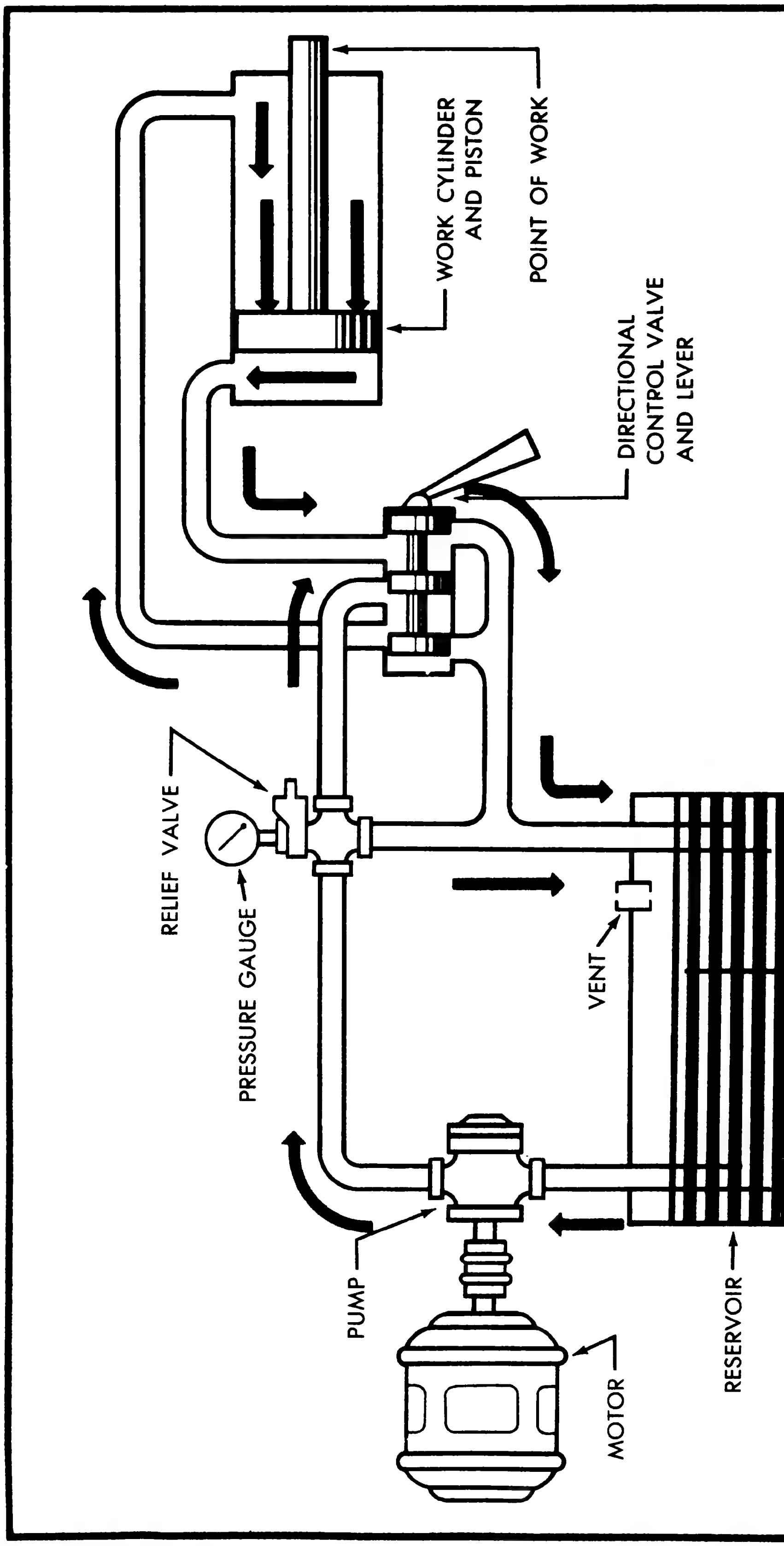

WORK CYLINDER AND PISTON
POINT OF WORK
DIRECTIONAL CONTROL VALVE AND LEVER
RELIEF VALVE
PRESSURE GAUGE
VENT
PUMP
MOTOR
RESERVOIR

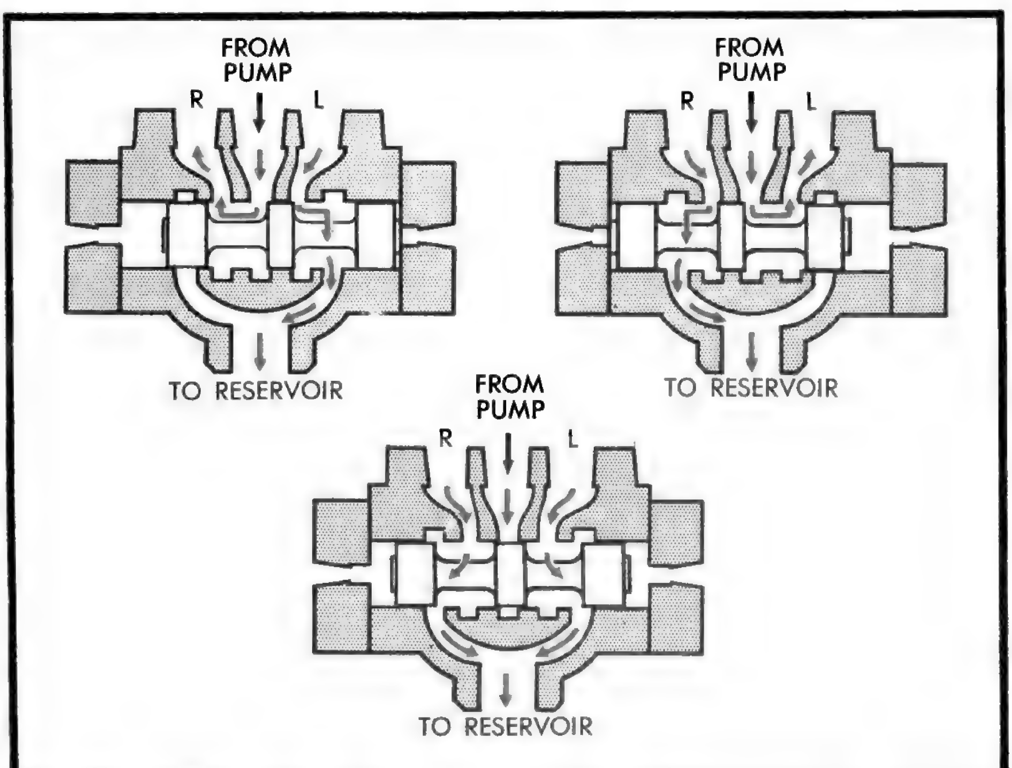
Figure 110

moving back and forth. Small grooves are therefore machined around the sliding surfaces of the spools, so that leaking liquid will encircle the spools and keep them lubricated and centered.

Four-way spool valves. Valves of this kind connect with four separate pipe lines. In a typical installation, such as the one shown in Figure 110, one line connects the valve to the pump. Another connects it with the reservoir, by means of two exits from the inside of the valve. The other two run to either end of the work cylinder.

As the piston of the four-way valve is moved back and forth, each end of the work cylinder is connected in turn with the pressure line from the pump, while the other end of the work cylinder is connected with the reservoir. This enables the work piston to be driven back and forth without meeting hydraulic resistance.

Figures 111–113 taken in conjunction with Figure 110 will show how a four-way valve works. The position of the piston in Figure 111 corresponds to the situation shown in Figure 110. With the piston of



Figures 111, 112, and 113

the valve to the far right in its cylinder, liquid from the pump flows to the right end *R* of a work cylinder, while liquid from the left end *L* of the work cylinder is being returned to the reservoir. In Figure 112, with the piston to the far left in its cylinder, relations are reversed, since liquid from the pump flows to the left end of the work cylinder, while liquid from the right end is being returned to the reservoir. In Figure 113, with the piston in an intermediate position, flow through the valve from the pump is shut off and both ends of the work cylinder can drain to the reservoir unless other valves are called on to control flow from them.

Since this valve is often used to drive a work piston back and forth after the manner described, it is sometimes called a reversing valve. The piston of the directional valve itself can be positioned in a number of ways: manually, as for example by a lever; mechanically,

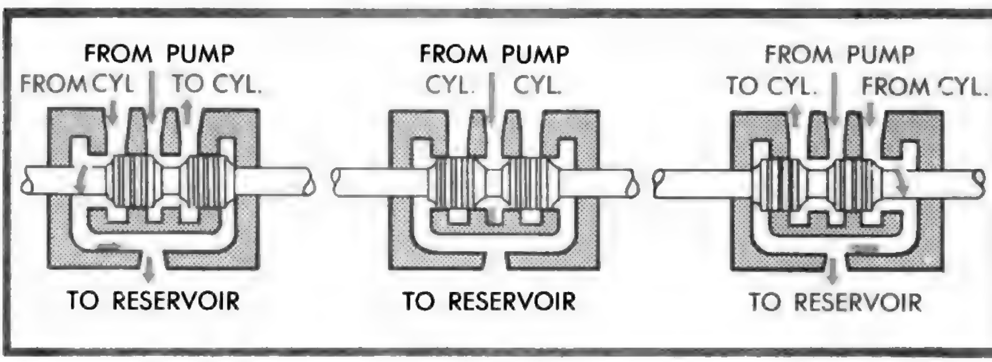


Figure 114

through the action of a cam brought into position at the proper moment by the work operation; mechanically by means of gears and shafting; electrically, in a number of ways; or hydraulically through the action of a pilot valve. Rotary directional valves are often used to position spool valves.

The valve in Figure 110 is operated by moving a lever. Figures 111–113 illustrate a four-way valve designed to be operated by a pilot valve. The pilot valve alternately delivers pressure to one end and then to the other of the cylinder of the four-way valve, in exactly the same manner as the reversing valve might be used to drive a work piston (see Figure 120). In Figure 111, for example, the pilot valve would stop delivering pressure to the left-hand end of the

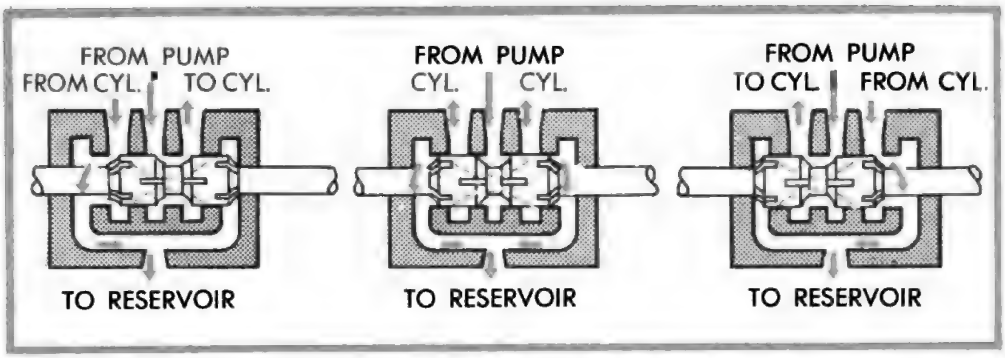


Figure 115

piston, and would begin delivering pressure to the right-hand end, thus reversing the position of the valve piston.

Closed and open center spool valves. In closed center valves the piston is solid and all passages through the valve are blocked when

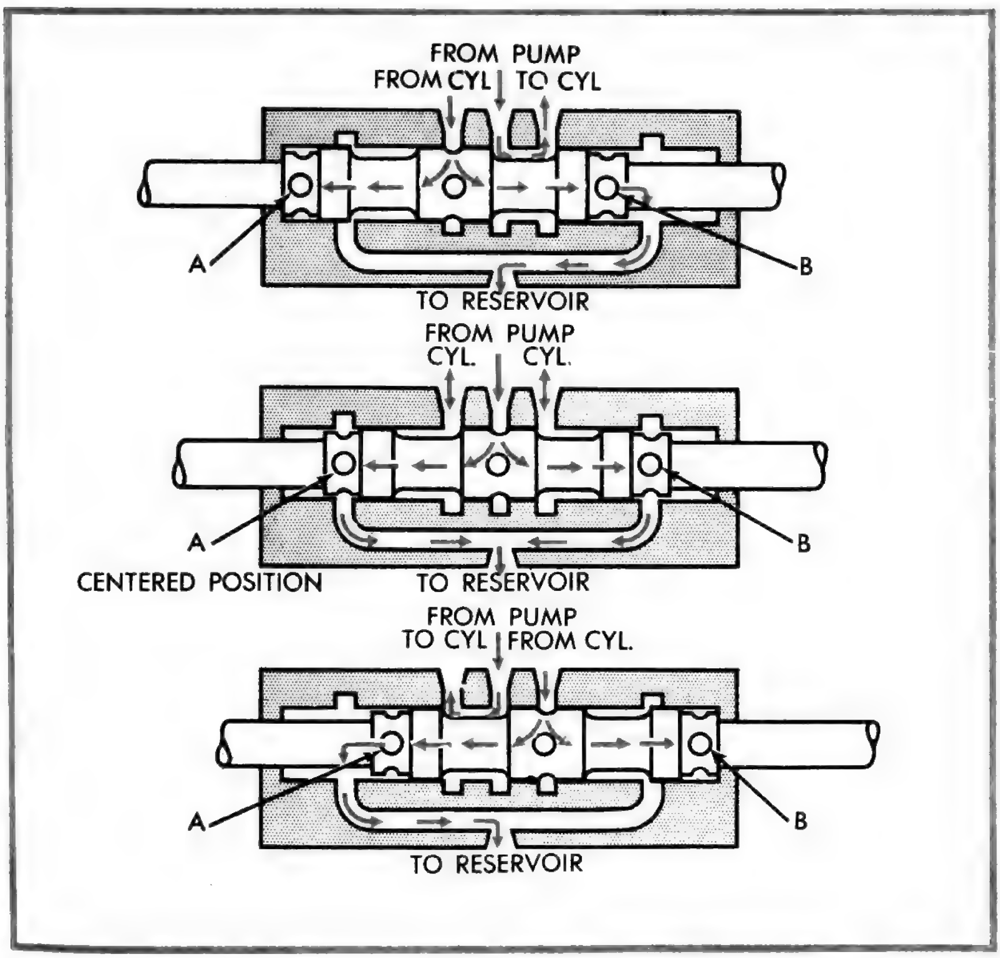


Figure 116

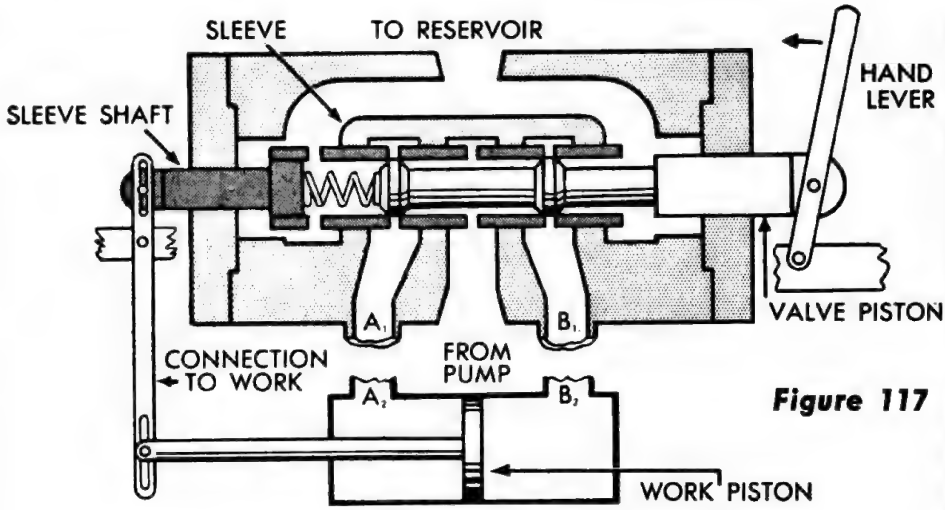


Figure 117

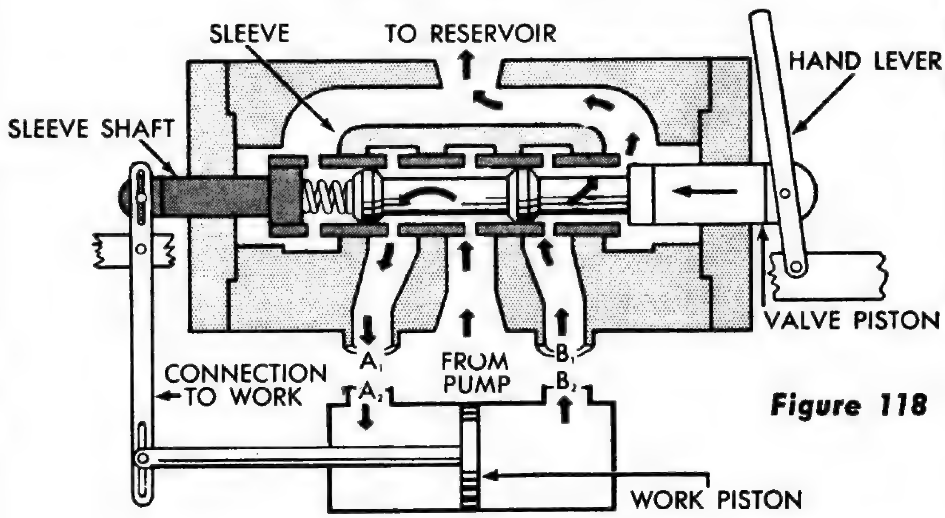


Figure 118

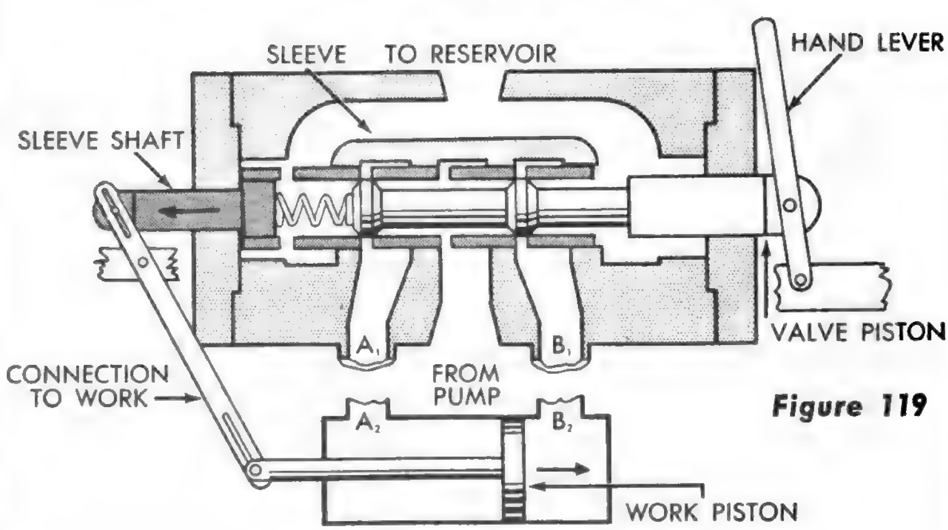


Figure 119

the piston is centered in its cylinder (Figure 114). In open center valves, the spools on the piston are slotted or channelled so that all passages are open to each other when the piston is centered (Figure 115). In some open center valves, passages to the work cylinder are blocked when the valve is centered, while liquid from the pump is carried in bared passages through the piston and out the other side of the valve to the reservoir (Figure 116). In valves of this kind liquid must be carried to both ends of the piston of the direction valve to keep it balanced—that is, to *A* and to *B* in Figure 116. Instead of discharging into the reservoir when the valve is centered, liquid may be directed to other valves so that a set of operations is performed in sequence.

Open center valves are used when the work cylinder does not have to be held in position by pressure, and where the power is used to perform a single operation. They also avoid shock to the system when the valve spool is moved from one position to another, since in the intermediate position pressure is temporarily relieved by the passage of liquid from the pump directly to the reservoir. Reversal of action is therefore smooth.

Closed center valves are used when a single pump or accumulator performs more than one operation, and where there must be no pressure loss in shifting the direction of stroke at the point of work.

Spool valves may be spring centered or spring offset, according to the position in which the piston of the valve stands when it is in neutral. A spring, that is, may return the spool to the center position whenever the force controlling it is released; or the spool may be returned to an off-center position on being released.

Follow valve. The cylinder wall of a spool valve can be constructed as a movable sleeve, and can be connected to a work operation in the manner indicated in Figures 117–119, so that motions of the valve piston precisely control motions at the point of work. Figure 117 diagrams this Vickers follow or "Servo" valve in neutral position.

Flow through the valve to either side of the work piston is blocked, since the piston heads of the valve close passages A_1 and B_1, which

lead to the left and right entrances A_2 and B_2 respectively of the work cylinder. A certain small amount of liquid is discharged by the pump through the valve when it stands in neutral position, so that the discharge pressure of the pump will not be sufficient to activate a relief valve protecting the system from overload (not shown).

Now if the valve piston is moved somewhat to the left by a similar movement of the hand lever, the pump will send liquid through A_1 to the left side of the work piston, while at the same time the valve will open up a passage from B_2 on the other side of the work piston, through B_1 and on to the reservoir (Figure 118). This will cause a related movement at the point of work, which in turn will shift the sleeve shaft of the follow valve to the left until the sleeve and pistons once more stand at neutral (Figure 119), but now somewhat farther to the left than they stood in Figure 117.

A valve of this kind can be used to steer a ship. When the steersman wishes to change course, he shifts the steering gear. This changes the position of the piston in the follow valve, and the work piston moves, altering the position of the rudder, which causes the valve sleeve to move until the valve is once more in neutral. No matter how the rudder may be positioned, it will be held steady, without strain upon the valve or the steering system.

Casualties with spool valves. The spools in these valves may become scored or coated with impurities from the liquid used. The action of the valve may become unbalanced due to wear on the spools or the body, so that the valve will work unsatisfactorily or stick. The valve may leak. The return spring may break.

Rotary Directional Valves

Valves of this kind are most frequently used as pilot valves to direct flow to other valves. A schematic view of a rotary directional valve used to control flow to a four-way spool valve is given in Figure 120. Liquid is passed from the pump through the rotary valve on to the

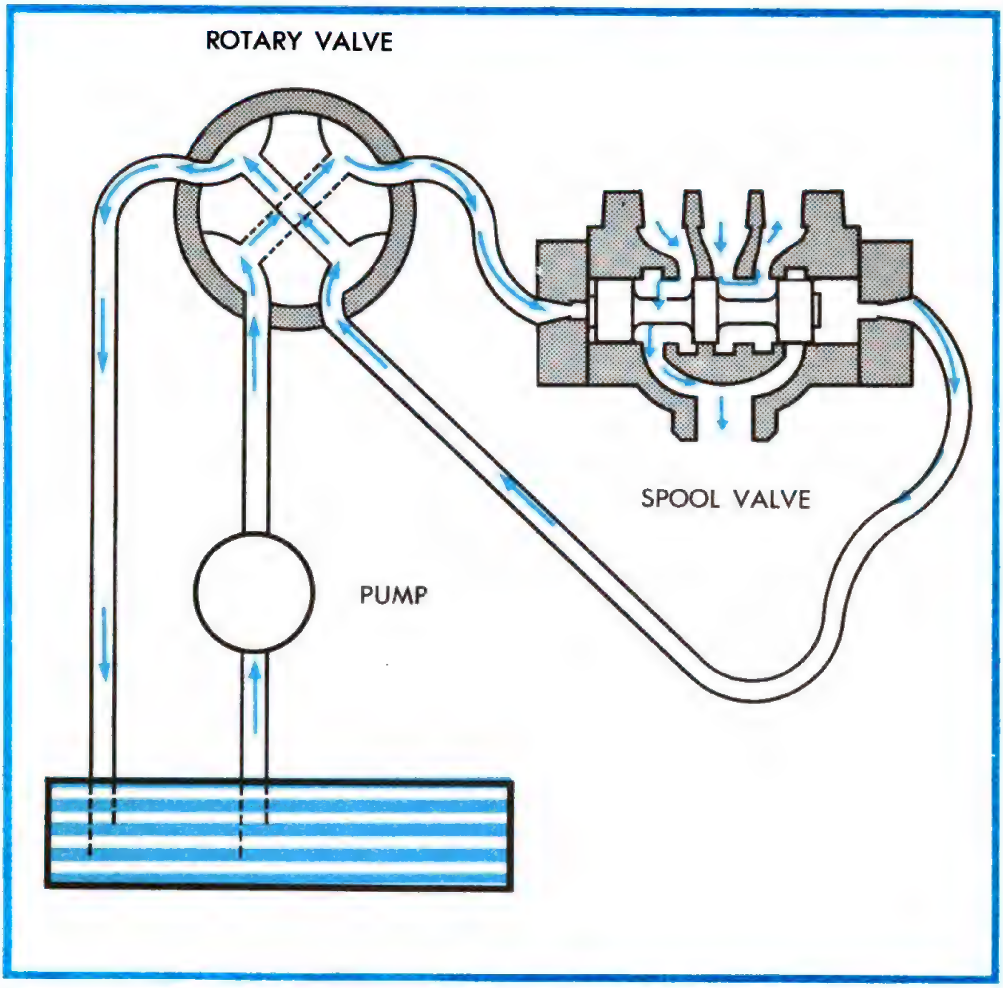


Figure 120

spool valve, where it positions the spool valve to direct flow from another pump to one side of a work piston (not shown).

Through a return line, liquid is carried from the other end of the spool valve through the rotary valve to the reservoir.

Figure 121 shows how a rotary type valve works. *A* and *C* show it set to deliver liquid through a spool valve to different sides of a work piston, while *B* shows the valve in neutral position, with all passages through the valve blocked. The valve shown is an open center model, since it contains two cross passages (one indicated by a dotted line in the figure). Other rotary valves are made with closed centers.

Rotary valves can be operated manually, mechanically, hydrau-

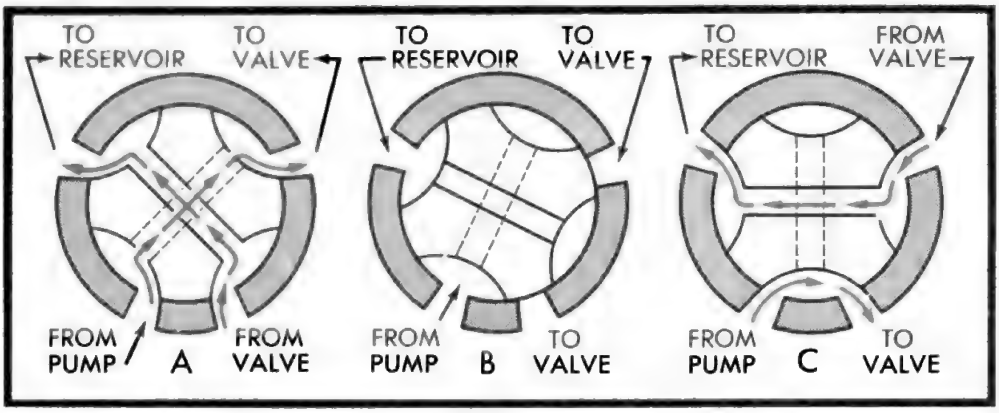


Figure 121

lically or electrically. They are not often encountered in ordnance hydraulic set-ups.

Flow Control Valves

Metering valve. A partially cutaway rotating core can be used to meter the flow of a liquid (Figure 122). As the position of the core is changed, a greater or smaller quantity of liquid is allowed to pass through the valve. In this way the speed with which an operating part moves can be controlled, the action of a valve delayed, or pistons cushioned at the end of a work stroke. The valve could be used to prevent a tool from breaking through too rapidly at the end phase of a work movement, or to bring a rammer to a stop at the end of its stroke without straining the system.

Time delay valve. Figure 123 diagrams the action of this valve. Liquid moving from the pump through a control valve (not shown) at different times takes three paths: (1) around the time delay valve past check valve *A* on to the point where work is to be done; (2) through the time delay valve; and (3) through check valve *B* to produce a force on the bottom of a piston in the time delay valve which is held in place by a spring. As an in-

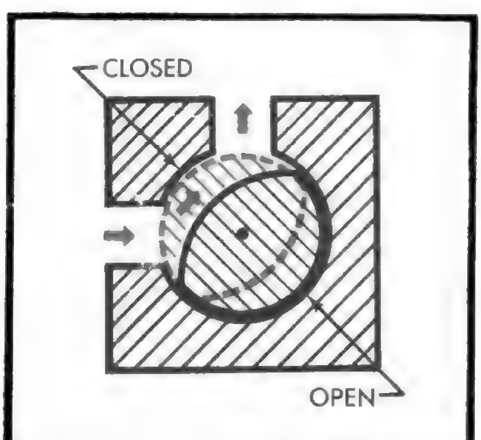


Figure 122

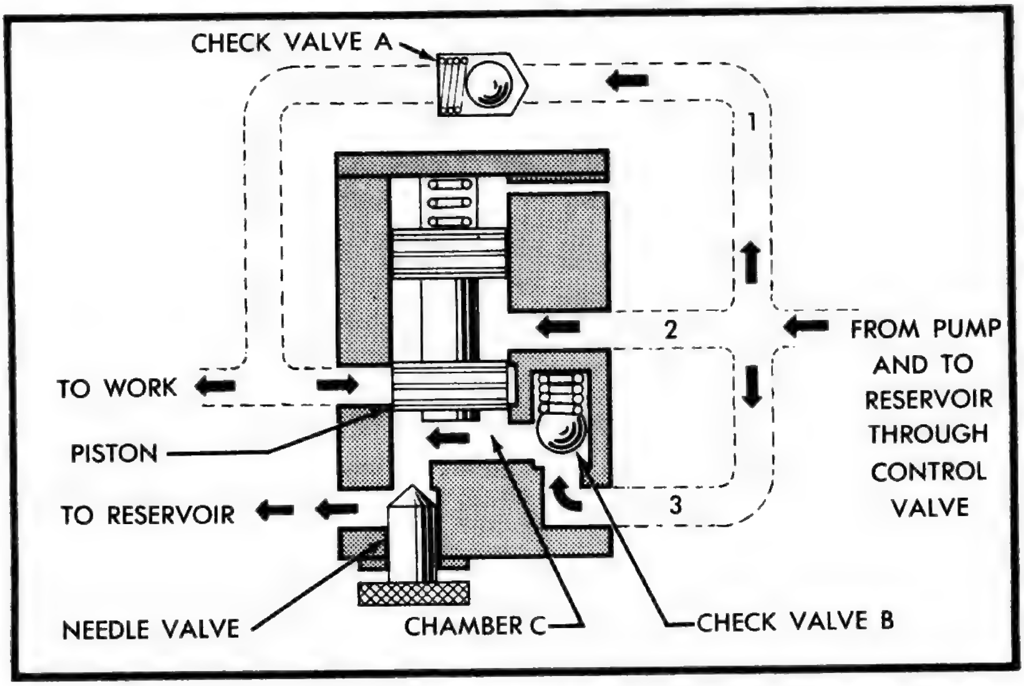


Figure 123

crease in hydraulic pressure compresses the spring of the piston, it rises to shut off flow through the time delay valve. Figure 123 shows the valve in this position.

When the position of the valve controlling flow from the pump is changed, liquid returns from the point of work, but is blocked from passage *1* around the time delay valve by the upper check valve, and from passage *2* through the time delay valve by the lower piston head of the valve. But a needle valve under the piston slowly drains liquid from chamber *C*, so that the piston can slowly descend, allowing liquid to pass over it and on to the reservoir by the same path that liquid from the pump originally entered the valve. This is made possible by a different positioning of a control valve.

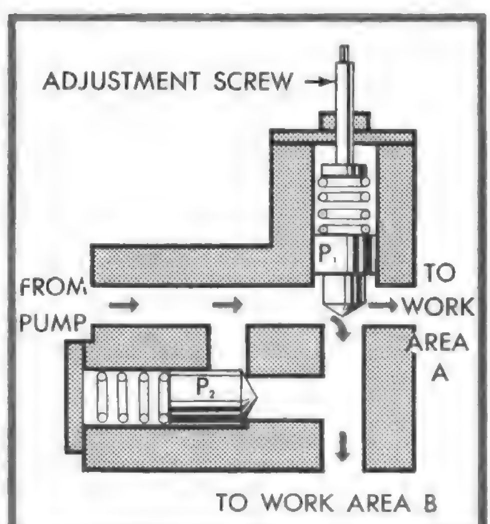


Figure 124

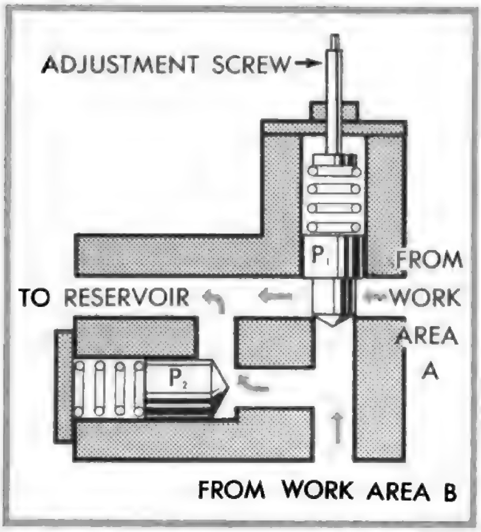


Figure 125

By-pass valve. Here flow runs in one passage until a certain pressure is reached. It is then also carried along a second passage to another work operation (Figures 124 and 125). Liquid flows around P_1 to one point of work until a pressure is reached which raises P_1 against the resistance of its spring. Then liquid will also flow through the second passage of the valve to do other work (Figure 124). P_1 is held open by a smaller pressure than is needed to open it, because of the greater area against which this pressure can then act. When the position of the control valve governing the by-pass valve is changed, liquid returns through the by-pass valve from work area A past the base of P_1, and from work area B past P_2 (Figure 125).

Foot valve. Here free flow is possible in one direction, while flow in the other direction is impossible until a certain pressure is reached.

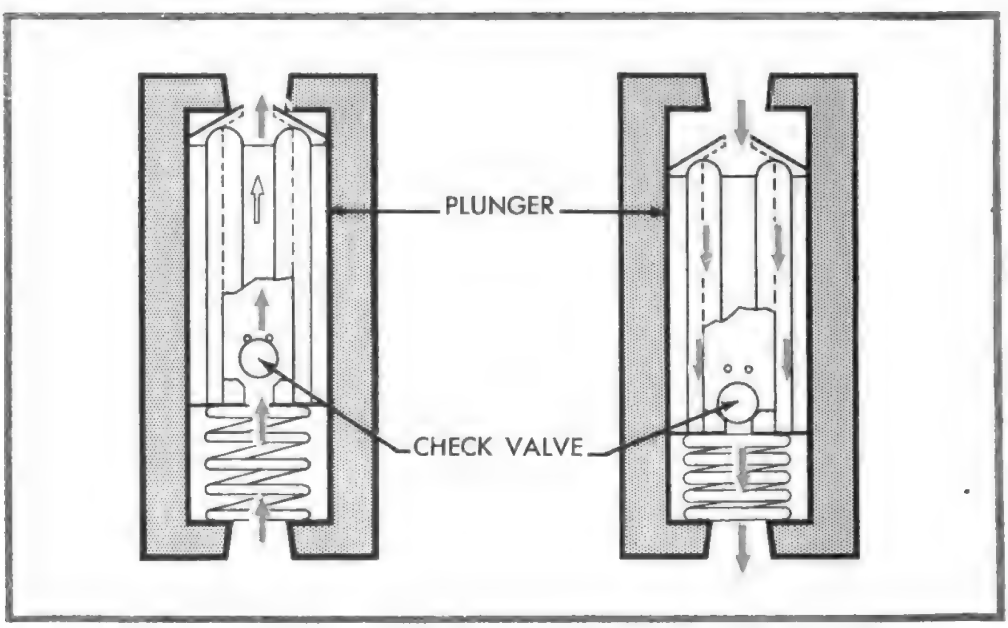


Figure 126

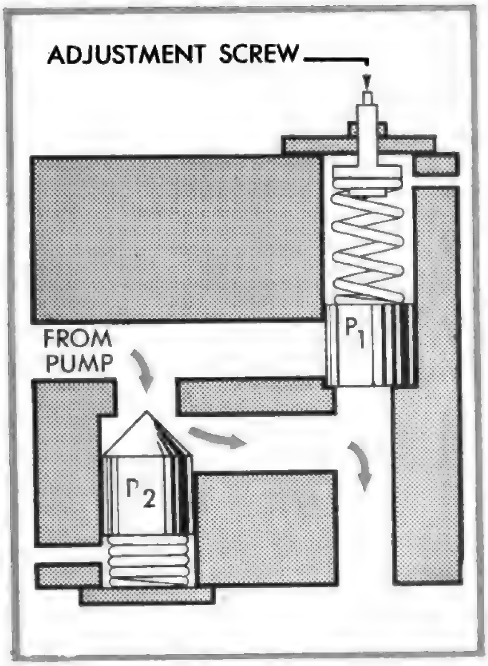


Figure 127

A nonadjustable foot valve is shown in Figure 126. The plunger is hollow and has an inner passage with a check valve at its base and an opening at the top, so that liquid can flow upward through the valve but is prevented from returning until pressure above the piston increases sufficiently to unseat the valve by compressing the spring at its base. Then the whole plunger descends, permitting downward flow until pressure above the valve is lowered to the resistance of the spring, when the plunger is once more seated.

The set-up for an adjustable foot valve is similar to that for the by-pass valve shown in Figures 124 and 125, except that the pistons are shaped and positioned to require restricted flow (Figures 127 and 128). Liquid moving from the pump to the point of work passes P_2 by compressing a light spring; while liquid returning to the reservoir goes past P_1 against the resistance of a spring set to compress at a definite pressure (Figure 128).

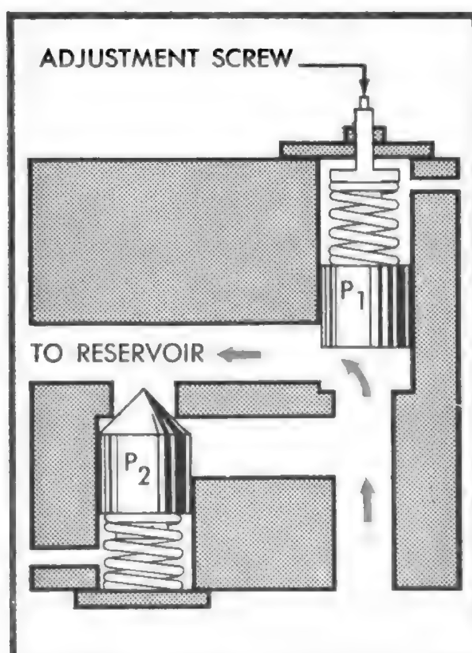


Figure 128

Prefill check valve. This valve permits liquid from a storage reservoir to fill the work cylinder of a hydraulic system during the first phase of the work action, when the work plunger is being moved rapidly to the point of work by gravity, or other means

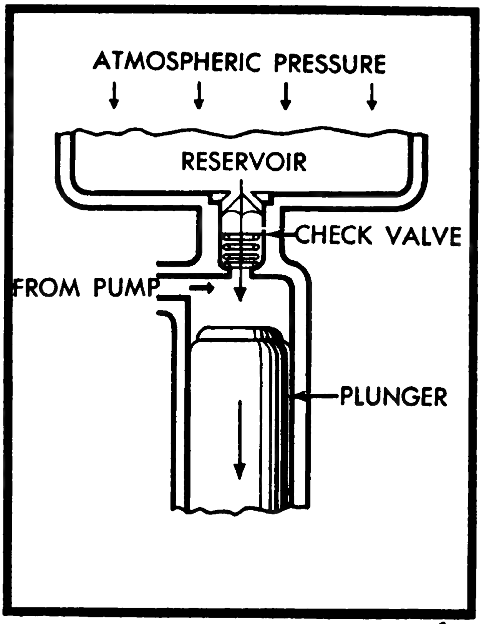


Figure 129

(Figure 129). In this way a small volume pump can be used to do the work, instead of the large pump that would be needed if all the liquid to fill the work cylinder had to be furnished by the pump.

In the surge type illustrated, the check valve is opened by atmospheric pressure acting on the free surface of the liquid in the reservoir when a partial vacuum is set up inside the work cylinder as the plunger descends. The check valve is closed by its spring when the plunger comes to rest, or when pressure in the work cylinder has risen sufficiently. From this point on, further movement of the work plunger is accomplished by pump action.

The check valve might also be opened by liquid directed to it by a pilot valve, and it might also be closed by that valve instead of by a spring.

Valve Blocks and Panels

The varied and delicately controlled operations performed by hydraulic systems today in industry and in the Navy often depend upon quite complicated combinations of pumps, valves, mechanical parts, and even electrical circuits. Such systems must be constructed so that they can be easily controlled, and so that they will give a minimum of trouble in operation.

Valve blocks. In many instances it is expedient to group a number of units together. Valves very closely involved in a series of operations are organized into a valve block, the different parts of which may be connected with each other not only hydraulically but

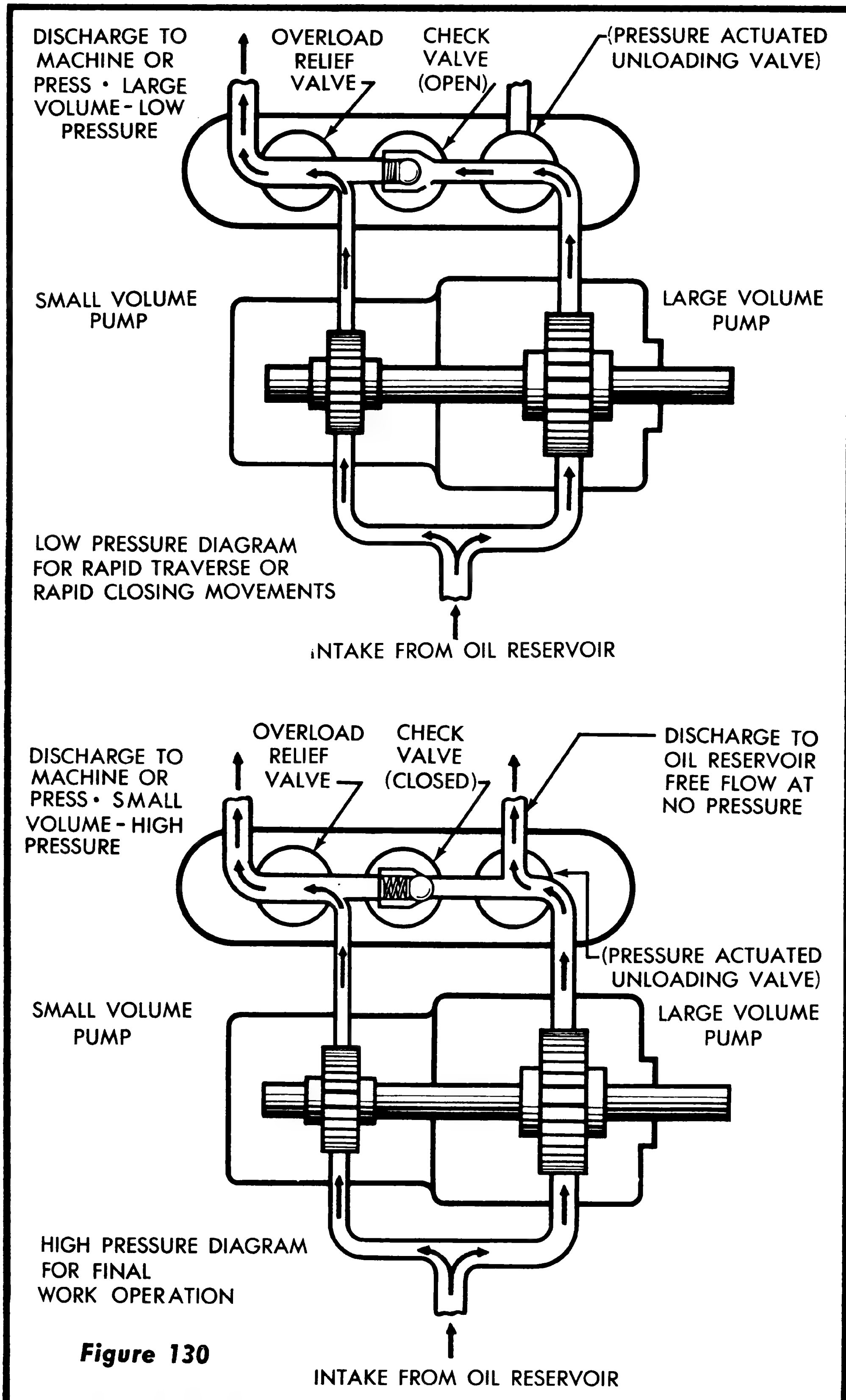

DISCHARGE TO MACHINE OR PRESS • LARGE VOLUME - LOW PRESSURE
OVERLOAD RELIEF VALVE
CHECK VALVE (OPEN)
(PRESSURE ACTUATED UNLOADING VALVE)
SMALL VOLUME PUMP
LARGE VOLUME PUMP
LOW PRESSURE DIAGRAM FOR RAPID TRAVERSE OR RAPID CLOSING MOVEMENTS
INTAKE FROM OIL RESERVOIR
OVERLOAD RELIEF VALVE
CHECK VALVE (CLOSED)
DISCHARGE TO MACHINE OR PRESS • SMALL VOLUME - HIGH PRESSURE
DISCHARGE TO OIL RESERVOIR FREE FLOW AT NO PRESSURE
(PRESSURE ACTUATED UNLOADING VALVE)
SMALL VOLUME PUMP
LARGE VOLUME PUMP
HIGH PRESSURE DIAGRAM FOR FINAL WORK OPERATION
INTAKE FROM OIL RESERVOIR

Figure 130

also mechanically and even electrically, so that the valves may be brought into position in any of these three ways.

The main valve block of the train and the elevation control systems of the 5-inch 38 gun, for example, consists of a half-dozen sliding valve or piston units that are so connected. Operation of the units composing the valve block is also coordinated with a considerable number of other units located elsewhere that are necessary to govern the action of the gun. The fluid used in the hydraulic operation of these valves is delivered at a number of pressures from pumps or from pressure reducing valves such as those shown in Figures 107 and 108. In addition, the whole system is organized so that it can be controlled in a number of ways, from complete operation at the gun by hand-power alone, to remote automatic control based on electrical signals from the gun director system.

Combining different work operations. In many industrial hydraulic installations, it is desirable to have a comparatively large volume of liquid available under relatively low pressure at one phase of an operation, as for example in the first stages of a movement, while later a smaller volume of liquid under higher pressure could better finish the task.

A number of solutions to this problem are possible. One arrangement makes use of a prefill check valve such as was shown in Figure 129. A second arrangement uses a large-volume low-pressure pump and a small-volume high-pressure pump working together to produce the result, as shown in Figure 130. Delivery of liquid from both pumps goes to the work area until a pressure is built up which is strong enough to open the unloading valve, when the high pressure from the small pump closes the check valve and liquid from the large pump is diverted to the reservoir. High-pressure delivery goes on continuously, but is diluted by low-pressure delivery when pressure in the system drops below the pressure setting of the unloading valve. The overload relief valve is set at the pressure above which it is not desired that the small-volume pump should deliver liquid.

Panels. Still other arrangements have been devised for combining

the results attainable with different volumes of delivery. Some of these systems are organized so that they can be controlled by *panels* like the one shown in Figure 131, where movement of the panel lever controls the action of the system for advance, coarse feed, fine feed, stop, and return, according to the setting given to the control dials on the face of the panel.

Panels can be used with either a single constant delivery pump, or with two constant delivery pumps—one delivering low pressure and one delivering high—or with a single variable delivery pump capable of supplying different volumes of liquid without changing its speed of operation. Variable delivery pumps will be described in Chapters 11 and 12.

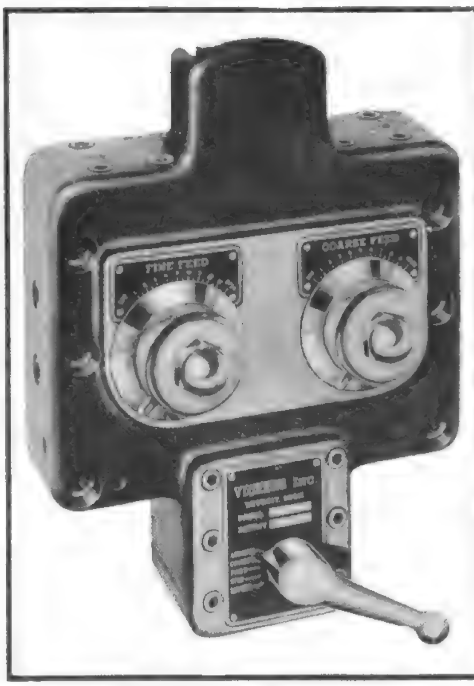


Figure 131

Through the use of hydraulic setups such as those we have been describing, extremely flexible control of movement becomes possible. Rotary or back-and-forth movement in either direction at any speed can be secured while constant power is being used. The operation can be *held* at any point, since any point in the operation can be made the neutral point of the hydraulic action. Complex series of movements can be integrated into a pattern of movements, so that almost any desired sequence of results can be secured.

Methods of Operating Valves

Valves are operated manually, mechanically, hydraulically or electrically, or by a combination of these methods. The mechanical-hydraulic combination is very common, as when spring pressure moves a part in one direction and liquid pressure provides the counter movement.

Manual operation. Gate, globe and needle valves are almost invariably operated by hand. Many valves or valve parts are set by hand, as for example the pilot valve part of a compound relief valve, although the pilot valve itself works by a combination of mechanical and hydraulic means. Some directional valves are hand operated. The dials of a panel system are hand set.

Mechanical operation. The mechanical parts used may be pivots, levers, cams, gears, cranks, springs, etc. Gravity is also used to operate valves, as for example the many check valves which close by gravity. Many valves close by spring action, as for example check valves and relief valves. Directional valves are often positioned by cams, gears, levers, etc.

Hydraulic operation. Check and relief valves open by means of hydraulic pressure. Directional valves may be positioned hydraulically. This means is always employed in operating amplifier valves. Air pressure is used to work air vents.

Electrical operation. Pressure switches can be used to make or break a circuit at a given hydraulic pressure. The switches can be set in a number of ways—as for example to reverse an action or start another process when the work has been moved a certain distance, or when pressure has reached a certain level. A solenoid or electromagnet is often used to position a valve, especially spool type directional valves. When a current flows through the solenoid, a core to which the valve is physically attached is drawn into the unit. When the current is turned off, a spring or hydraulic pressure returns the valve to its former position.

QUESTIONS

1. Distinguish between spool and rotary type directional valves.

2. Why are the spools of directional valves grooved around their circumference?

3. How can a four-way directional valve be used to drive a work piston back and forth?

4. Distinguish between closed and open center directional valves.

5. How are open center spool valves kept in hydraulic balance?

6. Explain how a follow valve can hold a work operation in neutral at any point in the operation.

7. What is meant by a metering valve?

8. Describe the action of a time delay valve.

9. What is the purpose of a prefill valve?

10. What is a valve block? What are panels?

11. In what ways are valves operated?

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Chapter 8

INTRODUCTION TO PUMPS

This chapter first discusses pumps in general, and then considers one type—reciprocating pumps. The general discussion deals with what a pump is; with the classification of pumps by types and groups; with head and energy relations in pumps; with the comparison of pump types; and with installation, maintenance and casualty problems.

What A Pump Is

We all know that pumps are used to lift or transport liquids, and are familiar with certain common kinds, as for example the simple reciprocating pump described in Chapter 1. But there are many kinds of pumps in general use, serving a variety of purposes, of which perhaps the most important is to transfer liquids for performing a work operation through use of their hydraulic properties (Figure 132).

A pump is a mechanism through which an external source of power is utilized to apply a force to a liquid. In Chapter 2 we learned about the five factors which govern the physical behavior of liquids, and it was pointed out that of these factors applied forces are by far the most important so far as naval applications of hydraulics are concerned. Almost without exception the practical method of applying a force to a liquid is to use a pump. The purpose may be

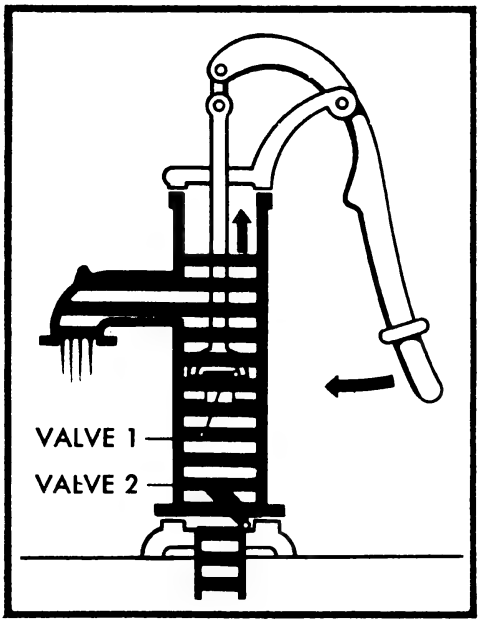

Figure 132

to raise the liquid from one level to another, as when lifting water from a well; to transport it through a pipe, as in an oil pipe line; to give it a high velocity, such as the stream from a fire hose; to move it against some resistance, as when filling a boiler under pressure; or to force it through a hydraulic system against various resistances, for the purpose of doing work at some point. In ordnance applications the last of these uses will be the most common one encountered, although all of the other applications can be found aboard ship.

A pump develops no power of its own. It simply transfers power from some source, as for example an electric motor, to a liquid. The pumping action takes place in a pump chamber, which is connected by an intake or suction pipe to a reservoir or other supply of liquid. The chamber has an outlet or discharge pipe to deliver the liquid pumped. At all times the general principles developed in Chapters 1 and 2, including the conservation of energy, will hold true, and by means of them every action of the liquid can be explained. This will be further amplified in the course of this chapter.

Classification of Pumps by Types

Pumps can be variously classified and grouped, depending on the purpose the classification is to serve. We shall find it useful to distinguish five types, which we shall then group according to the manner in which they discharge liquid.

1. *Liquid or gas operated pumps,* where a liquid or gas under pressure is *directly introduced* into the pump chamber to produce the pumping action.

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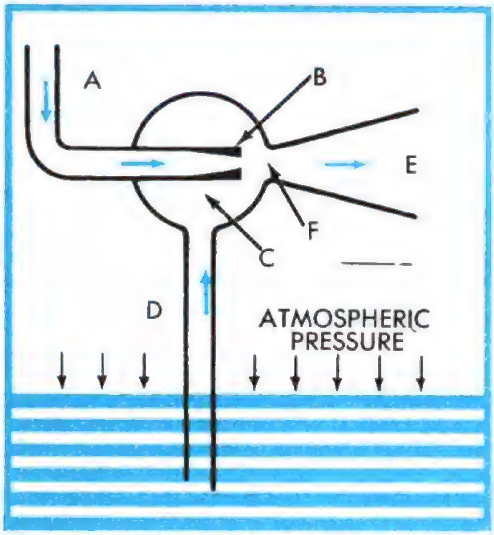

Figure 133

An example is the jet pump (Figure 133). If steam, or even liquid or a gas under pressure, is driven into a chamber having an exit, as it rushes out through the exit it will carry some of the contents of the chamber with it. This will lower the pressure in the chamber. If the chamber is connected to an open supply of liquid, a pumping action will be set up, since atmospheric pressure on the surface of the liquid will force some up into the inlet pipe and eventually into the chamber and out through the exit.

In Figure 133, steam or liquid under pressure enters chamber *C* through pipe *A*, which is fitted with a nozzle *B* having a reduced area so as to increase the velocity of the pressure jet. The contents of the chamber at *F*, in front of the nozzle—at first air and later liquid—are driven out of the pump through the discharge pipe *E*. The size of the discharge pipe is increased beyond the chamber in order to decrease the velocity of the discharge and thereby transform some of its velocity head into pressure. As the introduced steam or liquid knocks part of the contents of the chamber into the discharge, pressure in the chamber is lowered and atmospheric pressure on the supply surface forces liquid up through inlet *D* into the chamber and out through the discharge.

This pump contains no moving mechanical parts. The only motions involved are those of the pressure jet and of the liquid being pumped, the jet acting directly on the liquid. The principle of this pump is a valuable one to understand because of its extreme simplicity. Any handyman with a few pipe fittings and a wrench can rig up a jury pump which will do work wherever liquid or gas pressure is available.

A common example of a jet pump is a flit gun or a paint spray gun. Pumps of this kind are sometimes used to feed water into

boilers, as auxiliaries to the regular feed pumps. Boiler steam supplies the pumping energy. There are other kinds of liquid or gas operated pumps, but this type will not be further discussed in this book since they are seldom used in the Navy.

2. *Reciprocating pumps*—pumps involving *a back and forth motion of mechanical parts.*

In Figure 134, when piston *P* moves to the left, on the suction stroke, the pressure in cylinder *C* is decreased below the atmospheric pressure acting on the liquid in the reservoir. Atmospheric pressure, therefore, forces liquid up the suction pipe, through check valve *2* and into the cylinder. Atmospheric pressure acting on the liquid in the discharge pipe and the head of liquid standing there closes check valve *1*. When the piston moves to the right, on the discharge stroke, the increased pressure in the cylinder closes valve *2* and opens valve *1*, so that liquid is forced up into the discharge. This process repeated many times can build up a considerable head in the discharge end of the system. The hand pump shown in Figure 132 is a reciprocating pump acting vertically.

3. *Rotary pumps.* Here a rotary motion carries the liquid from suction to discharge on a path which may be circular, elliptical, or follow some other curve.

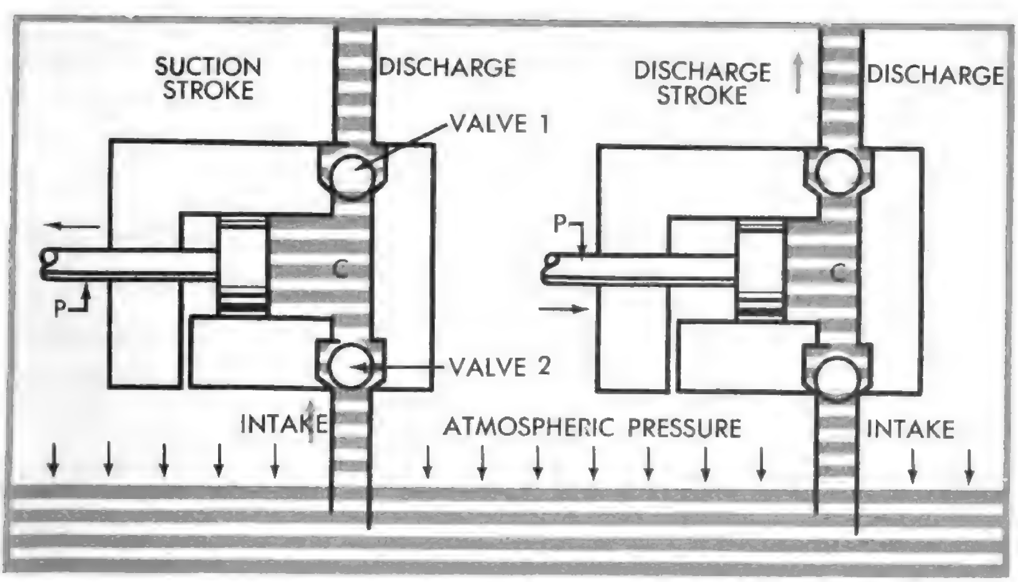

Figure 134

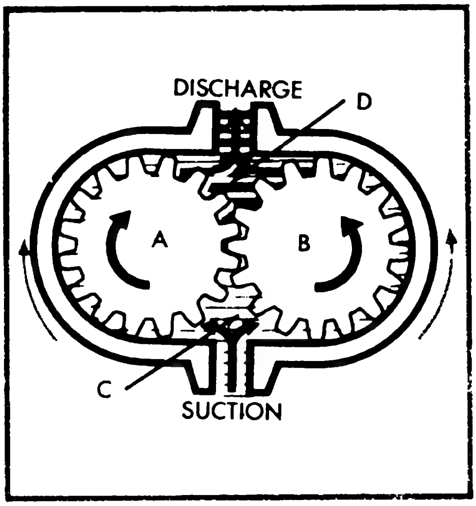

Figure 135

Rotary pumps offer great scope for invention, since the variety of possible arrangements is almost endless. The principle according to which they all operate is indicated in Figure 135. Gears *A* and *B* rotate in opposite directions, one gear usually being driven by the other in the particular kind of pump diagrammed. Liquid enters the pump at the bottom, and is carried up to the discharge in the small spaces formed between the gear teeth and the wall of the pump chamber. As a volume of liquid enters one of these spaces, a pair of teeth also disengages at the center of the pump. Pressure in chamber *C* is decreased, because a volume of liquid has been taken from the chamber, while simultaneously a space empty of liquid has entered the chamber, due to the unmeshing of the gear teeth at the center of the pump. As a result a volume of liquid is forced up through the suction pipe by atmospheric pressure on the free surface of the source of supply. When each volume of liquid caught between the teeth reaches chamber *D*, it is prevented from returning to chamber *C* by the meshing of the gears at the center of the pump chamber. The gears form a continuous seal there. Pressure in chamber *D* is therefore increased, so that liquid is forced out of the pump through the discharge pipe.

Rotary pumps are almost universally used to pump lubricating oil in automobile engines. They can be constructed with many kinds of rotating members, some of which are not gears, strictly speaking, since one of them cannot be driven by the other. Various kinds of rotary pumps will be described in Chapter 10.

4. *Centrifugal pumps,* where in addition to the effect produced by carrying the liquid around and out of a chamber, advantage is taken of the *centrifugal force* generated by rotation.

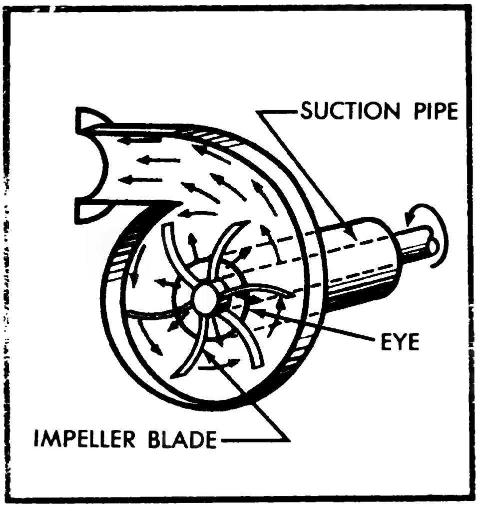

Figure 136

Centrifugal force is the force generated by rotation which acts directly outward from the center of rotation. This will be fully explained in Chapter 9, when centrifugal pumps are discussed.

In practically all centrifugal pumps the liquid enters the pump at the center through the *eye* (Figure 136), and is given a rotary motion in the pump chamber by the rotation of a number of blades (the impeller). The rotation of the impeller in a true centrifugal pump does two things to the liquid. Centrifugal force drives the liquid directly outward from the center, setting up a greater pressure at the outer edge of the chamber than at the eye. At the same time the liquid is also pushed around and around in the pump by the turning of the blades, and is given more and more velocity as it moves farther out from the eye. The liquid finally escapes into the discharge. By gradually widening the discharge pipe and thereby reducing the velocity of the liquid, most of the velocity head produced by centrifugal pumps is transformed into pressure head. In this form it is more available for doing work.

The impeller blades of most centrifugal pumps are curved, the nature of the curve and the dimensions of the pump chamber and the discharge pipe being determined by the kind of flow desired. These questions are very complicated, and since they concern the designer almost exclusively they will not be considered further in this book. It should be noted that the direction of rotation of the impeller shown in Figure 136 is counter-clockwise, so that the liquid is pushed around the chamber by the blades, rather than being carried in them. This is the usual arrangement.

Centrifugal pumps are almost universally used in the water cooling systems of automobiles. Various kinds are described in Chapter 9.

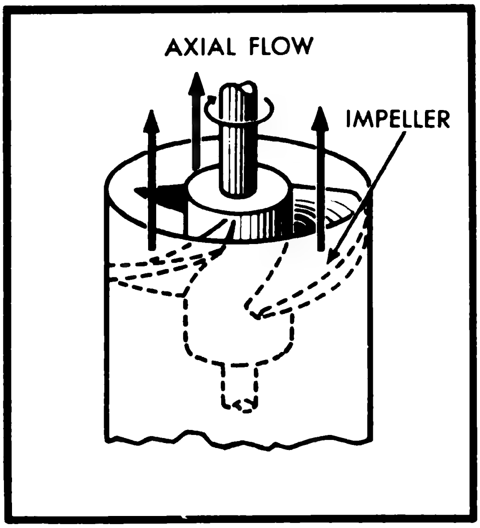

Figure 137

5. *Propeller pumps.* Here the blades are set so that they merely *sweep* the liquid out of the pump by their rotation.

The blades in a propeller pump are set at an angle to the shaft, somewhat after the manner of the blades of a ship's propeller. The action of the pump is exactly that of a common ventilating fan surrounded by a pipe. Figure 137 shows a shaft and impeller from a propeller pump. The liquid leaves the impeller in a direction parallel to the axis of the shaft. In centrifugal pumps the liquid leaves the pump in a direction at right angles to the axis of rotation.

In *mixed flow pumps,* the actions of centrifugal and propeller pumps are combined in various proportions by altering the shape and angle of the impeller blades and the shape of the pump chamber. In the ideal centrifugal pump, the liquid is given a rotary motion and swung out of the pump, whereas in the ideal propeller pump the rotation sweeps the liquid out of the pump. The impeller of a mixed flow pump is designed to combine these two actions. The blades stand somewhere between a position parallel to the drive shaft and at right angles to it. Propeller and mixed flow pumps are discussed in more detail in Chapter 9.

Classification of Pumps by Groups

Pumps can be grouped according to whether they discharge liquid in volumes separated by a period of no discharge (positive displacement), or in a continuous flow (non-positive displacement). In addition, pumps can be classified according to whether or not the quantity of liquid discharged by the pump can be varied without changing the speed of rotation of the source of power (variable or constant delivery).

Positive and non-positive displacement pumps. A positive displacement pump is one in which a definite volume of liquid is delivered for each cycle of pump operation, regardless of the resistance offered, provided the capacity of the power unit driving the pump is not exceeded. A non-positive displacement pump is one in which the volume of liquid delivered for each cycle is dependent upon the resistance offered to flow. This type of pump produces a force on the liquid that is constant for each particular speed of the pump. Resistance in the discharge line produces a force in the opposite direction. When these forces are equal, the liquid is in a state of equilibrium and does not flow.

If the outlet of a positive displacement pump is completely closed, either the unit driving the pump will be stalled or something will break. If the same thing is done to a non-positive displacement pump, the discharge pressure will rise to a maximum for that type of pump operating at that speed. Nothing more will happen except that the pump will churn the liquid and produce heat.

Positive displacement pumps deliver liquid in separate volumes, with no delivery in between, although a pump having many chambers may have an overlapping delivery which minimizes this effect. Non-positive displacement pumps deliver a practically continuous even flow for any given set of conditions of speed and resistance.

For every back and forth motion of the piston in the pump shown in Figure 134, a complete cylinder of liquid will be delivered to the outlet while the piston is moving to the right on the discharge stroke. Nothing at all will be delivered while the piston is moving to the left, on its suction stroke. If the outlet were completely blocked, as by a shut-off valve, it would be impossible to move the piston to the right because of the virtual incompressibility of the liquid filling the chamber. If sufficient force were applied to the piston, the pressure in the chamber would rise until finally something would break.

Obviously this single chamber pump would give a badly pulsating delivery. In the rotary pump shown in Figure 135, where each of the spaces between the teeth of the gears is a pump chamber, delivery

will overlap so that a more nearly constant flow will be obtained. Another method of reducing the pulsation is to provide an air chamber near the pump outlet to damp the pulsations, but there is always some pulsatory effect with positive displacement pumps. Pulsations are objectionable because they cause jerky movements in driven mechanisms and set up vibrations in the whole system.

For the centrifugal pump of Figure 136, the liquid is not broken up into isolated volumes to be delivered one by one, but a pressure differential is created in the solid stream by the rotary and sweeping action of the impeller so that a smooth continuous flow results.

Reciprocating and rotary pumps of necessity are positive displacement pumps, while jet, centrifugal, propeller and mixed flow pumps of necessity are non-positive displacement pumps.

Constant and variable delivery pumps. All pumps will deliver liquids at different volume rates if run at different speeds. It is not practical to vary the speed with ordinary positive displacement pumps, although these pumps can be especially constructed to deliver different volumes of liquid while being run at the same speed. They can be constructed, that is, to give constant or variable delivery for each cycle so that volumes of delivery can be obtained without changing the operating speed of the pump.

This is managed with reciprocating pumps by altering the length of the work stroke of the piston. Thus in Figure 134, each discharge stroke of the piston might displace 200 cubic inches of liquid, but the distance the piston travels might be shortened to give a stroke which would displace only 180 cubic inches. This would cut the volume of delivery 10 per cent without altering the speed.

Gear type rotary pumps are not normally constructed to give variable delivery at constant speeds. Other kinds of rotary pumps have been developed, however, to achieve this result. They will be considered in Chapters 11 and 12.

Centrifugal pumps deliver different volumes of liquid while running at a constant speed, since the volume of discharge varies with

the head against which the liquid must be discharged. It is not practicable, however, to use this fact to control volume of delivery under a constant head.

Head and Energy Relations in Pump Systems

General considerations. No matter where or how pumps are used in hydraulic systems, all head and energy relations will conform to the laws of hydraulics and of the conservation of energy already discussed in Chapters 1 and 2. Like all other machines, pumps effect only energy transformations—they do not create energy. The energy applied to a liquid by means of a pump goes either into the production of usable pressure or velocity in the liquid, or into friction losses. The energy thereby released in the system precisely equals the input energy supplied to the system.

The presentation of energy relations in a pump system, however, leads in detail to diagrams of a somewhat different appearance from those previously discussed. There are three reasons for this:

1. The pump may stand *above*, *at* or *below* the surface level of the source of supply. If it stands on the same level as the free surface of the supply, a diagram of the system raises no new considerations, since the pump with its source of power acts on the liquid like any other applied force. If the pump stands *below* this level, however, as shown in Figure 138, a certain amount of energy in the form of gravity head will already be available when the liquid enters the pump. If the pump stands *above* this level, as shown in Figure 139, a certain amount of energy is needed simply to get the liquid into the pump.

2. By means of pumps, energy from a power source is often introduced into a system at some midpoint, whereas in Chapter 2 we analyzed situations in which energy was applied to a system at one end of the system.

3. Many pumps operate in closed systems in which liquid from a reservoir circulates around through the system and back again to the reservoir by virtue of the pump action. This is the case, for

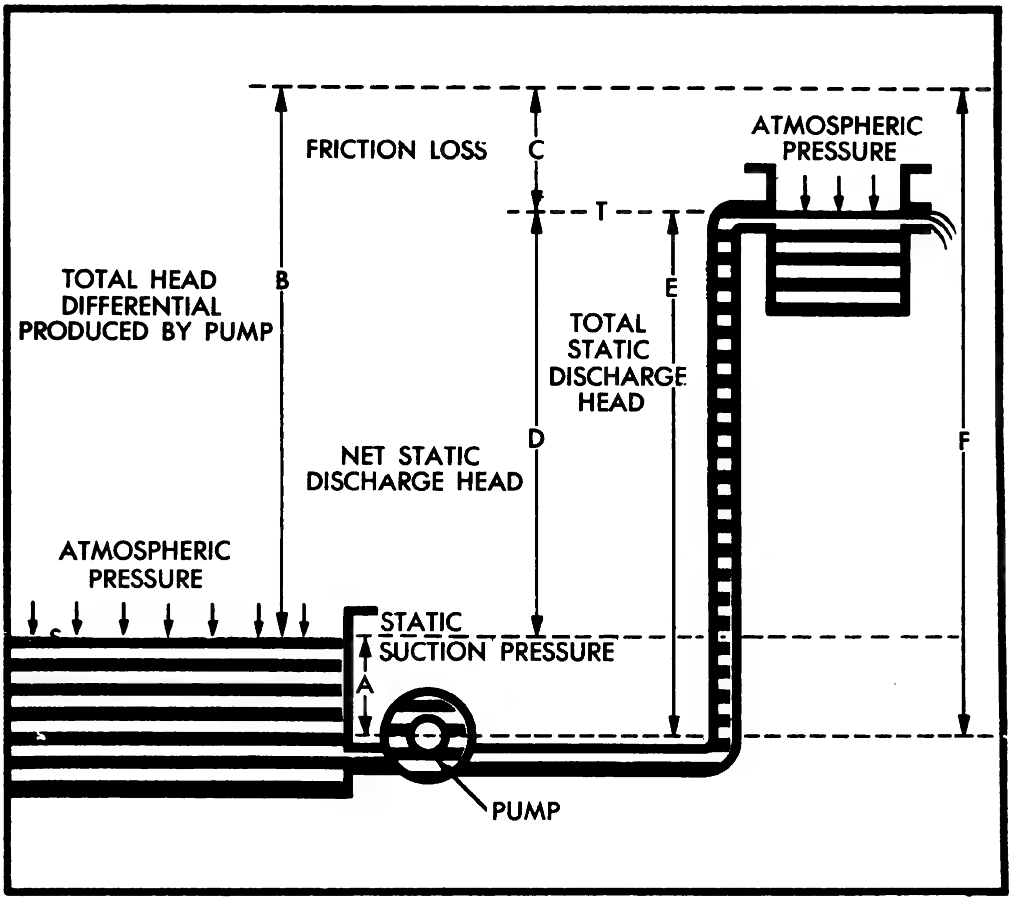

Figure 138

example, with cooling systems, as well as most systems designed to perform a work operation.

Head relations with the liquid under pressure at the inlet. When the pump stands at the vertical distance *A* below the free surface *S* of the source of supply, as shown in Figure 138, there is a static pressure head *A* on the suction side of the pump. This head will be a part of the total input head necessary to produce the required output head *F*. The action of the pump produces the total head differential *B*, which can be broken down into friction loss *C* and net static discharge head *D*. *D* is the vertical distance above *S* that the energy supplied to the system through the pump has raised the liquid. *E*, the total static discharge head, is the vertical distance from the center of the pump to the surface of the liquid in the discharge reservoir, and is equal to *D* plus *A*.

Put a little differently, the system contains two input factors, A and B. A enters the system as initial elevation or potential head, B as a result of the energy supplied to the pump. On the output side we have C and E, or friction loss and total static discharge head. Input and output precisely balance, since $A + B = C + E = F$.

It should be noted, in contrast with the cases considered in Chapter 2, that no mention is made of velocity head. This is because we are considering the overall problem of moving stationary liquid at level S to stationary liquid at level T. The liquid will of course have a velocity head at any point in the pump or the pipes, but since the liquid ultimately comes to rest again within the system considered, all of this velocity head will be converted either into potential head or into friction, or turbulent loss.

Atmospheric pressure will, of course, act upon the free surface of

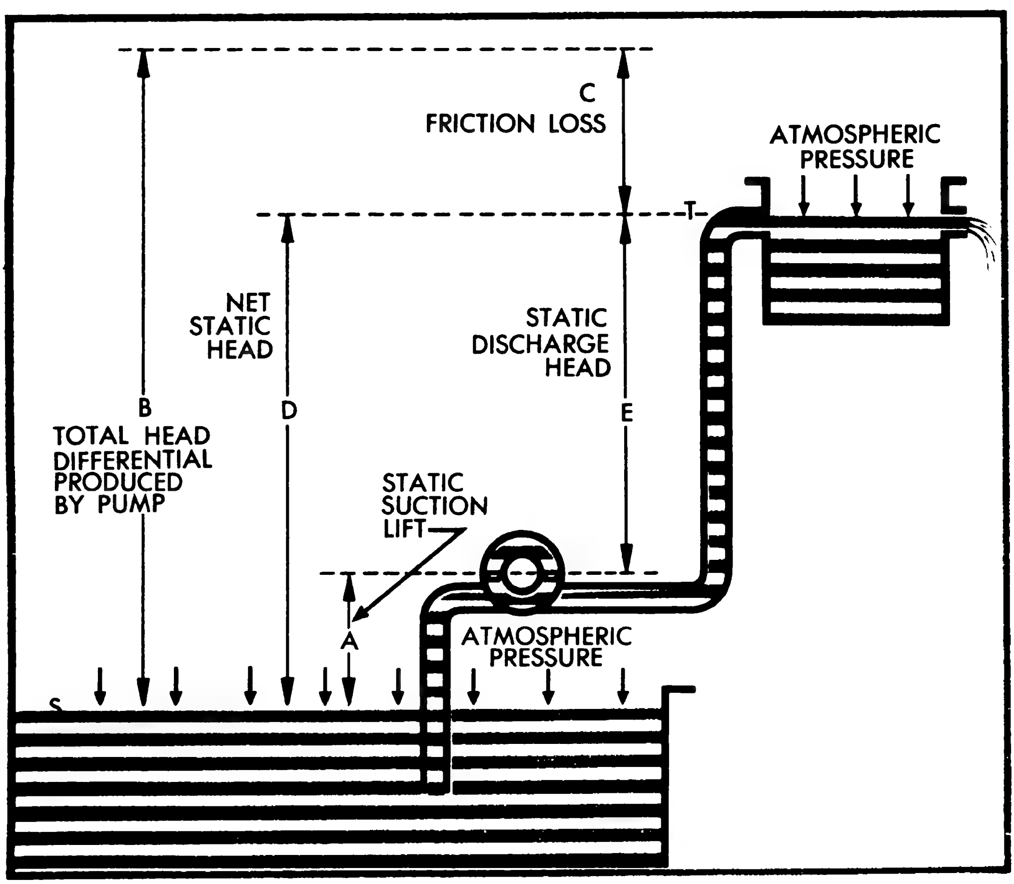

Figure 139

the liquid at S and at T, but since it creates equal heads on opposite sides of the pumps, it cancels out.

Head relations with the pump inlet under suction. In Figure 139 the pump stands the vertical distance A above the free surface S of the source of supply. Energy must therefore be applied to the liquid to get it into the pump. In addition, enough energy must be applied to produce static discharge head E, plus C which is lost in friction, if the liquid is to be raised to the level T.

B, the total head differential produced by pump action, is here the total energy input. It is divided into A on the suction side of the pump—the head consumed in raising the liquid into the pump—and C and E on the discharge side, which represent respectively friction head and static discharge head: $B = C + E + A$.

Velocity head again does not appear because the liquid comes to

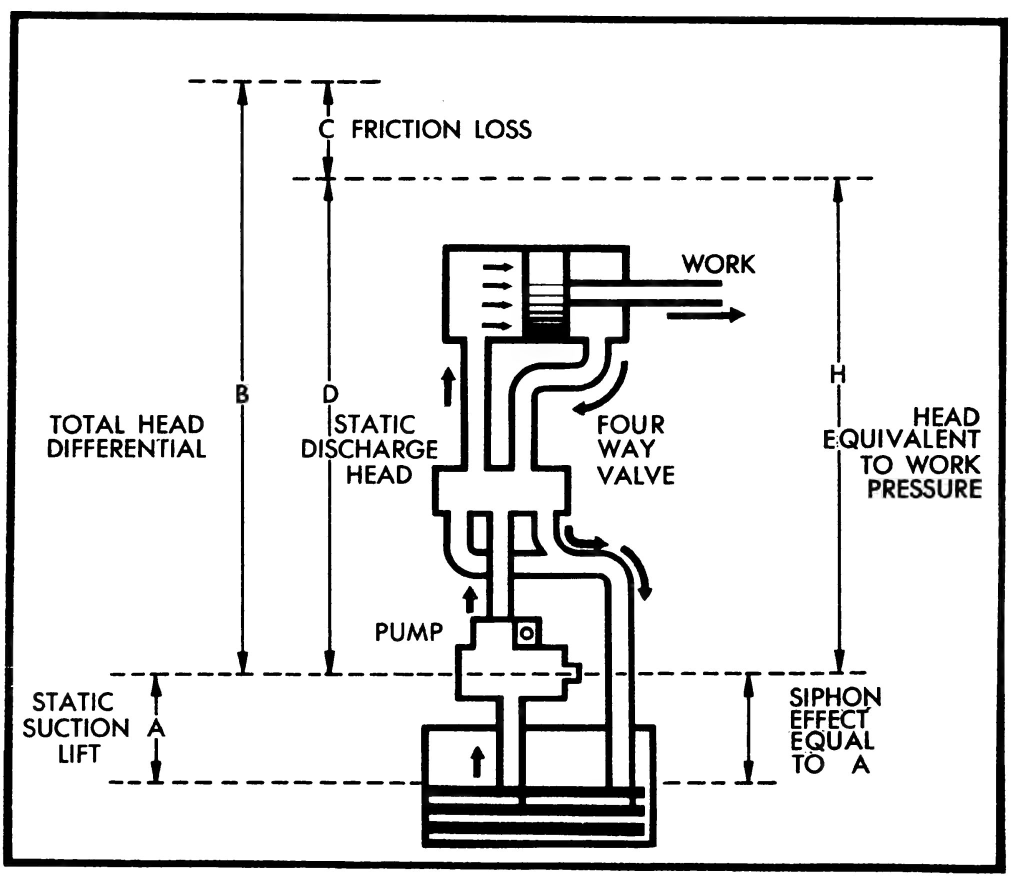

Figure 140

rest at the end of the cycle. Atmospheric pressure cancels out as before, only in this case the atmospheric pressure at S was necessary to lift the liquid up to the pump.

Head relations in a closed system. The situation here is indicated in Figure 140, where a pump is driving a work piston back and forth in a cylinder. To do this against the resistance offered we shall assume that it must develop a pressure equivalent to head H. Under this assumption, the total head differential B must be produced by the pump after the system has begun to operate. This is divided into friction head C, consumed in moving the liquid through the system; and static discharge head D, which is equal to H, and therefore produces the pressure required to do the work: $B = C + D$, and $D = H$, which we assumed was necessary for the work operation.

In this case, since the liquid returns to its original level and the system is closed throughout, there will be a siphon effect in the return pipe which will exactly balance the static suction lift A. Atmospheric pressure will play a part in the operation of the system only when it is being started up, and until the entire active part of the system is filled with liquid.

It is not necessary for the parts of this system to be located as shown in Figure 140. The work area, for example, does not have to be located over the pump. In many ordnance systems the pump is located in the reservoir, where it is submerged in the inactive liquid of the system. At whatever levels the different parts of the system may stand, the levels will be in balance with the system in operation.

Here again, since we start and finish with stationary liquid, whatever velocity head may exist at various points along the way will ultimately disappear into its equivalent head either as static pressure or as friction.

Energy relations. In Chapter 1 it was pointed out that energy or work is the product of a force multiplied by the distance through which it acts. This principle applied to liquids gives us the equiv-

alent rule that energy is equal to the volume of liquid moved, multiplied by the head against which it is pumped. Therefore, in each of the foregoing illustrations for a given volume of liquid the energy relations will be the same as the head relations.

Factors determining suction and discharge heads. As has already been pointed out, in open systems suction head is strictly limited by atmospheric pressure. This is also true of any siphon effect in a closed system if the reservoir is open to the atmosphere. This means that the lift of pumps at sea level will be greater than at higher altitudes, since the lower atmospheric pressure at higher altitudes will be balanced by a shorter column of liquid.

If an attempt is made to lift a liquid more than the distance atmospheric pressure is capable of raising it, the liquid will vaporize. This is because the lowered pressure above the liquid lowers the temperature at which it will vaporize. At a certain suction lift the liquid vaporizes even if quite cool, and nothing but vapor reaches the pump.

This also explains why the maximum suction lift for hot liquids is considerably less than for cold. Volatile liquids are also difficult to pump, since even a slight pumping action tends to vaporize them. Finally, suction lift depends on the density of the liquid. It takes a longer column of a light liquid than of a heavy one to balance atmospheric pressure at a given altitude. Thus for oil with a specific gravity of 0.881, the theoretical limit of lift at sea level would be 34/0.881 or about 38½ feet. In other words, it would take a 38½-foot column of this oil to balance atmospheric pressure at sea level, as compared with a 34-foot column of water, since volume for volume this oil weighs less than water in the ratio of 0.881 to 1.0.

Pumps of the liquid or gas operated, reciprocating and rotary types are able to draw up liquid if they are started empty, whereas centrifugal pumps must be primed before they will draw. This is because ordinary centrifugal pumps are not tight enough to pump air, and therefore the pressure above the liquid in the suction line will

not decrease sufficiently to allow atmospheric pressure to force the liquid up into the pump.

Pumps can be primed by pouring liquid into the pumps by hand; through use of a bypass connection from the discharge side of the pump; by putting a foot valve in the suction pipe to hold liquid there (see Figures 126–128); or by exhausting the pump of air with a vacuum pump.

On the discharge side, heads of almost any magnitude can be obtained. Pumps can be arranged to act in stages, so that each pump adds more pressure to that already supplied by the others. With centrifugal pumps the volume of liquid pumped will vary with the pressure, even with constant speed of operation. There is a maximum pressure attainable for any given speed of operation and density of liquid being pumped, beyond which no liquid will flow. For positive displacement pumps this is not true; the volume depends only upon speed of operation regardless of pressure, within the capacity of the source of power.

"Characteristics" of a pump. The pressure developed by a pump or the volume of liquid it pumps will change if the load against which it acts or the pump's speed of operation are changed. In addition, pumps of different types will operate differently under similar conditions. The effect of discharge pressure on pump output is very different, for example, as between positive and non-positive displacement pumps. The behavior of a pump under varying conditions is shown in curves, known as the *characteristic curves* of the pump, which show:

1. The relation between head produced and rate of discharge when the pump is running at a constant speed. This curve would show the gallons per minute discharged at different heads or pressures (*A* in Figure 141).

2. The horsepower which must be supplied to the pump to get different rates of discharge against different heads (*B* in Figure 141).

3. The *efficiency* of the pump, or the usable energy output divided

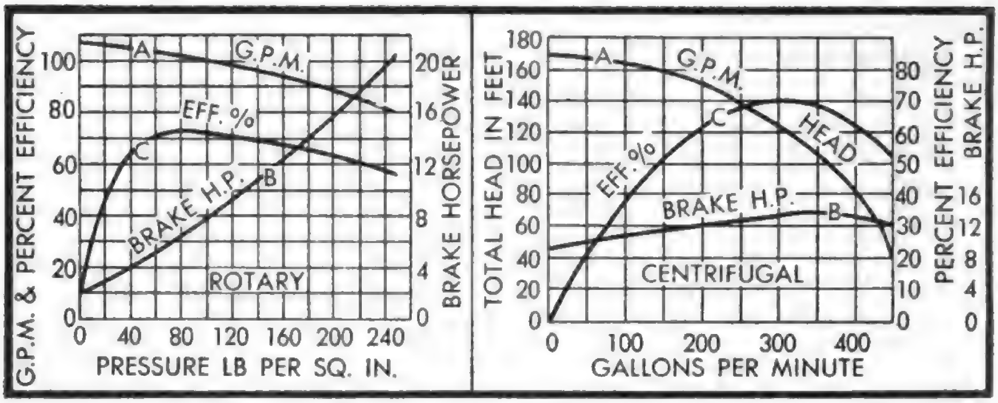

Figure 141

by the energy input when the pump is run at different rates of discharge. Here the work that could be done by the liquid pumped is compared with the work required to pump the liquid (*C* in Figure 141). The manner in which pump efficiencies are calculated lies beyond the scope of this book.

Typical curves. Each pump has certain definite characteristics that can conveniently be shown graphically. Figure 141 gives typical curves for a rotary and for a centrifugal pump with speed of pump operation constant. Rotary pumps as a rule run more efficiently at lower heads. Thus in the rotary pump diagrammed, maximum efficiency was reached at a pressure of about 80 pounds per square inch (curve *C*). The power necessary to run a rotary pump increases almost directly as does the pressure produced by the pump at constant speed. This is shown in the diagram by the fact that the brake horsepower curve *B* is almost a straight line. There is also relatively little difference in the rate of discharge for different pressures produced. This is shown by the fact that the gallons per minute curve *A* does not depart far from the horizontal. The departure shown is due solely to leakage and not to any inherent characteristics of the pump design.

With most centrifugal pumps, on the other hand, the rate of discharge drops quickly as discharge pressure increases. This is shown in the centrifugal pump diagram by the fact that output equals about 350 gallons per minute against a head of 40 feet, but drops almost to zero against a head of 160 feet (curve *A*). Centrifugal

pumps run most efficiently for most designs at higher rates of discharge. The pump diagrammed is shown to run most efficiently at a discharge of about 300 gallons per minute (curve *C*) as compared with about 100 gallons per minute for the rotary pump (combination of curves *C* and *A*). Relatively less input energy per output is needed, up to a limit, as the rate of discharge is increased (curve *B*).

Sometimes centrifugal pumps are spoken of as having a rising or a falling characteristic. In a pump with a rising characteristic, the head increases up to a certain point as the pump is run at a constant speed and the discharge is increased from zero (Figure 142). In a pump with a falling characteristic, the head continuously decreases under constant speed as the discharge is increased from zero. A pump with a flat characteristic shows little variation, up to a certain point, as the rate of discharge is increased. These differences are accomplished by details of design outside the scope of this book.

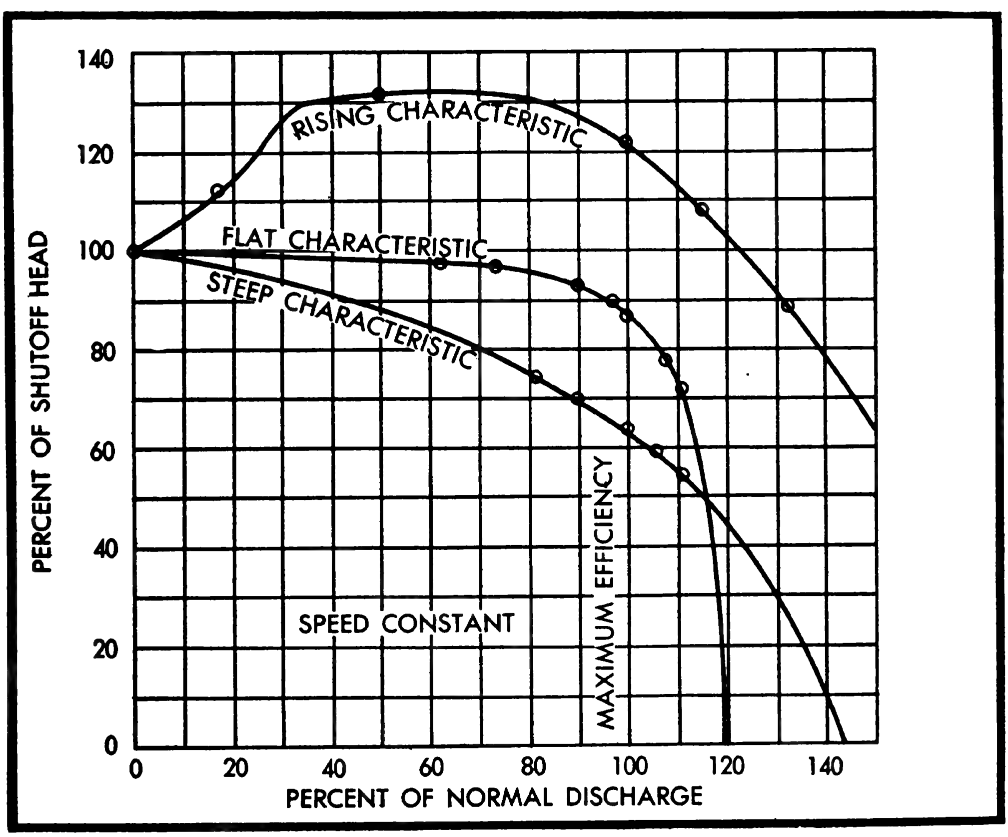

Figure 142

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Comparison of Pump Types

General. Pumps can be compared on a number of grounds, as for example:

Conditions of service:

Suction head to which they are adapted.

Volume and pressure of delivery.

Smoothness of delivery.

Range of pressure and volume (flexibility).

Dependability.

Type of liquid to be handled.

Conditions of use, inside and outside the system, with respect to temperature, etc.

(Much of the above can be derived from a study of curves showing the characteristics of the pumps being compared.)

Space occupied, and weight.

Power supply used—steam, electricity, etc.—considered both in relation to the power available and the manner in which it is used, with special attention to the question of speed control.

Initial cost; upkeep, replacement cost; casualties; overall economy of operation.

Specific comparisons: Relations to source of power. Reciprocating pumps were in general use when the customary motive power was the ordinary steam engine. The back and forth motion of the steam piston is similar to that of the hydraulic piston, and it is, in fact, possible to combine the two pistons as one functioning unit. The fact that the two units are adapted to move at about the same speed makes them good partners. Today, however, reciprocating pumps are often driven by electric motors.

Centrifugal pumps came to the fore with the introduction of turbines and electric motors, which were capable of much greater speeds and were naturally adapted to rotary motion. In recent years, rotary pumps of many kinds have been developed to handle situations requiring extreme flexibility of operation. Some of them can be adjusted to give the variations in liquid displacement required to perform many special operations.

Conditions of flow. Both reciprocating and rotary pumps can supply a constant volume of liquid against widely varying pressures, if the source of power is provided with a control or governor to keep it from racing or slowing down with load changes. The pressure they can operate against is limited by the source of power, whereas the pressure which can be developed by centrifugal pumps is a complicated function of speed, rate of flow and the head against which they work. For very large heads, therefore, positive displacement pumps are far superior to centrifugal pumps.

Positive displacement pumps are also more efficient when small volumes are required, and also if head and discharge must vary widely at different times. In other words they have a flatter characteristic than centrifugal pumps. Rotary pumps need to be specially designed to operate against high pressures.

Pulsations in flow. As already noted, positive displacement pumps give a pulsating flow, the reciprocating type being more pulsatory than some kinds of rotary pumps. Pulsations send pressure surges through the system, causing unsteady action, vibration of parts, water hammer, and metal fatigue, so that metal parts are less able to resist strain. The pulsations can be minimized by timing the operation of valves, by including an air chamber on the discharge side of the pump to soften the shock, or by building up pressure in an accumulator to smooth out variations. Arrangements with reciprocating pumps to minimize pulsations will be described in the second part of this chapter.

Liquids pumped. Most rotary pumps are not adapted to liquids carrying abrasives which may wear out the rotating elements and cause leakage, although the rotary plunger type is admirably suited to such conditions (see Chapter 10). Rotary pumps do not agitate the liquid as severely as centrifugal pumps. This may be a factor when air can enter the system and cause unsatisfactory pump action. They are excellent pumps for thick liquids. Pumps which are lubricated by the liquid being pumped should not be used with liquids having small lubricating power, as for example water. This includes most rotary pumps.

Structural considerations. Centrifugal pumps have relatively fewer moving parts than reciprocating pumps, and no valves. While reciprocating pumps when new are usually more efficient than centrifugal pumps, the latter retain their efficiency longer. Most rotary pumps are also without valves, but they have closely meshing parts between which high pressures may be set up after they begin to wear. Liquid under pressure may get into these spaces of small clearance as the pump elements rotate, and set up pressures there which are opposed to pump action. As will be pointed out in Chapter 10, some rotary pumps are constructed to eliminate back pressure.

Size and space factors. Vertical reciprocating pumps can be mounted on bulkheads, thus saving floor space. In general, centrifugal pumps can be made much smaller than reciprocating pumps giving the same result. There is an exception, in that positive displacement pumps delivering small volumes at high heads are smaller than equivalent centrifugal pumps.

Cost. Centrifugal pumps cost less when first purchased than other comparable pumps. The original outlay may be as little as one-third or one-half that of a reciprocating pump suitable for the same purpose. The cost of operation depends on comparative power costs as between any electricity and steam, and on upkeep costs. Since the capacity of a reciprocating pump varies directly as its speed of operation, the higher speeds of present day reciprocating pumps have given them a somewhat more favorable cost position than they had formerly.

Installation, Maintenance and Casualties

Proper alignment. It is important in the highest degree that pumps be lined up true with their driving mechanisms if they are directly connected. This means that the distance between the faces of the coupling joining them should be the same all the way around, and that the center line of each shaft should be a continuation of the center line of the other. If a pump is out of alignment, friction losses can be considerable, and parts can become so severely worn

as to be out of balance and subject to undue leakage, still further reducing efficiency. Pumps should be given a solid foundation, so that alignment will be maintained, and should be placed so that parts can easily be removed for repair.

No strain should be placed on the pump in setting it up. Both the intake and discharge pipes should be well supported so that the pump will not have to carry their weight. Piping should not be forced into position.

Avoiding air in the system. A properly installed pump of the right size and type for the work which is being used within its rated speed, will cause difficulty if air gets into the intake side of the system. It may mix with the liquid to cause an emulsion that can be pumped only spasmodically or with difficulty, or it may lead to air lock entirely preventing pump action. This is especially true with centrifugal pumps.

The suction approach to a pump should be one or two sizes larger than the pump connection, and there should be no bends near the pump. The approach should be short, direct, air-tight, and clean. The pipe should slant up towards the pump rather than down. Exception to this rule is when pumping hot liquid, in which case the piping should be slanted downward to the pump, so that the

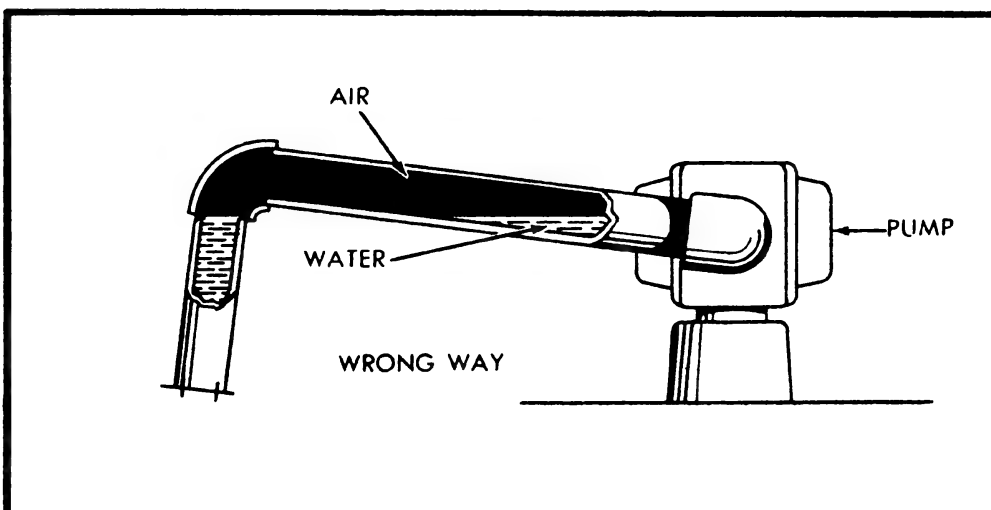

Figure 143

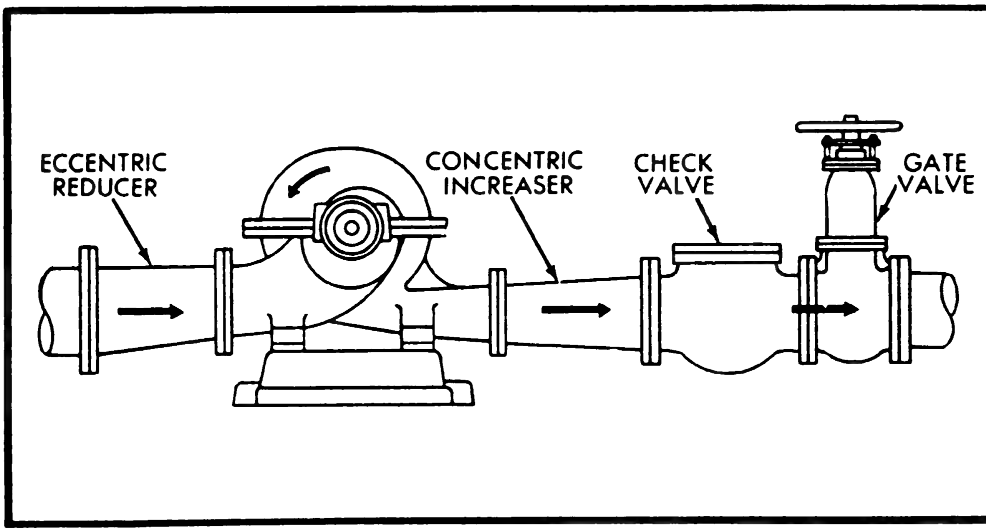

Figure 144

vapors will not enter the pump. Figure 143 shows how a down-slanting pipe in a suction line can interfere with flow by allowing an air pocket to form there. An eccentric reducer should be used in the suction line if the pipe does not slant upwards to the pump. As is clear from Figure 144, an eccentric reducer is a large-to-small pipe fitting with the two openings aligned on one side.

On the discharge side it is desirable in most instances to install an increaser (see Figure 144), so that some of the velocity produced by the pump will be transformed into pressure; a check valve, to prevent back flow into the pump and possible strain on the pump from system pressure when the pump is not working; and a gate valve to permit closing of the discharge. Installation of a gate valve is all the more advisable with centrifugal pumps so that they can be started with the discharge closed to prevent overload on the source of power.

Maintenance. When a pump is running smoothly, without overheating or vibration, it is presumably in satisfactory condition. It needs no special attention, beyond occasional lubrication of its bearings, and care to insure internal cleanliness. Foreign matter in a pump, due for example to dirty liquid, may seriously damage it, and in any event can hinder satisfactory operation.

All idle pumps should be turned by hand once daily, and by power

once a week. All valves—check, gate and relief, especially the latter—should be tested from time to time. Bearings should be lubricated at regular intervals.

Casualties. The following are indications that something is radically wrong with a pump. They should be investigated immediately:

Unusual noise.
Vibration.
Jerky action.
Failure to deliver liquid at the proper capacity or pressure.
Excessive leakage.
Overheating.
Overloading the prime mover.

For full instructions on the operation, maintenance, and repair of all types of pumps used in the Navy, Chapter 47 of the *Bureau of Ships Manual* should be consulted. The troubles most likely to be encountered with particular pumps will be discussed when each type is under consideration.

Reciprocating Pumps

In pumps of this type the source of power drives a piston or plunger back and forth in a cylinder. This action pushes liquid out into the discharge on the discharge stroke, and draws liquid into the cylinder from the source of supply on the suction stroke. For each stroke the same quantity of liquid enters and leaves the cylinder. The action is controlled by two check valves, one in the suction line and the other in the discharge line (Figures 132 and 134).

Reciprocating pumps were the first pumps to be widely used in modern hydraulic systems. They were at first usually driven by a steam engine, which was connected to the pump by a beam or other external mechanical linkage, so that the pump and steam pistons moved back and forth, stroke for stroke. In the old fire engine, the beam was often worked up and down by hand, as in Figure 145.

Direct acting pump. Although these early reciprocating pumps were bulky and inefficient according to the standards of today, they were

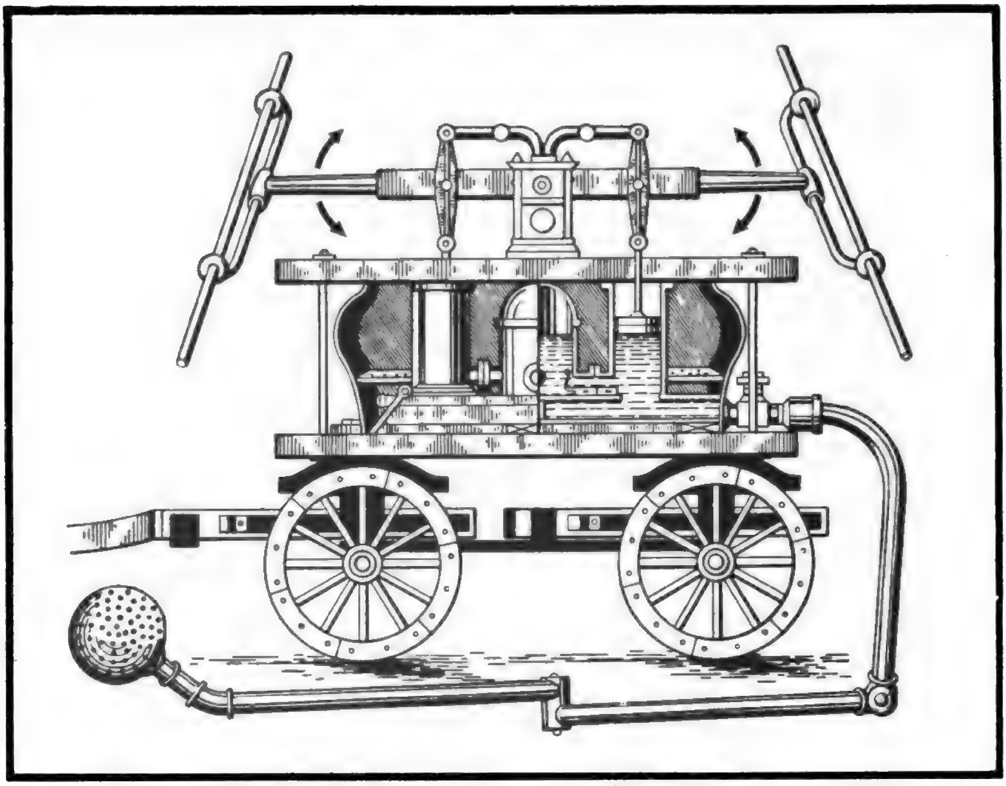

Figure 145

a great improvement on other early pumping devices. They were used both in water supply systems and to supply power to a hydraulic system for work operations. The invention of the direct acting reciprocating pump by Henry R. Worthington in 1840, however, opened up many new work applications. These pumps are arranged so that the pistons of the steam engine and the pump are directly

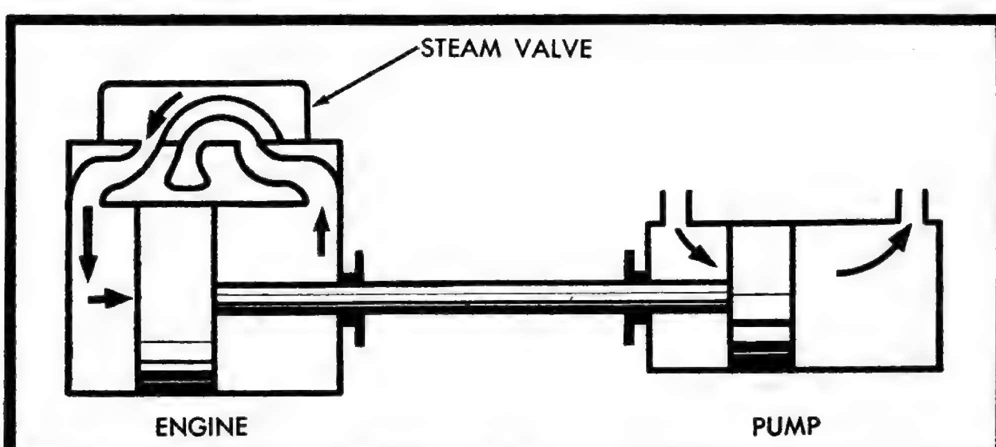

Figure 146

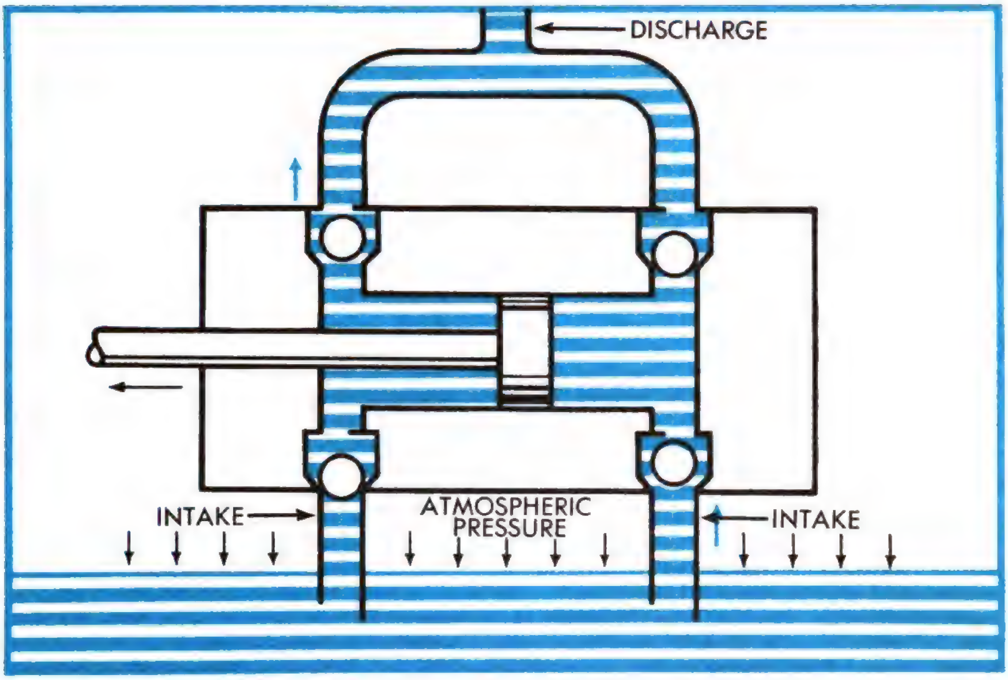

Figure 147

connected along a single straight line. They move back and forth together, the piston rod of the engine being also the piston rod of the pump, as indicated in Figure 145. Navy reciprocating pumps are almost exclusively direct acting when they are driven by steam.

Single and double acting pumps. The pump shown in Figure 134 is single acting. It discharges when the pump piston is moving in but one direction—to the right in the drawing. If the arrangement of pipes and valves to the right of the piston were duplicated at the left-hand end of the cylinder, as shown in Figure 147, the pump would become double acting. Liquid would then continuously be entering and leaving the cylinder from one end or the other. The two inlets might be

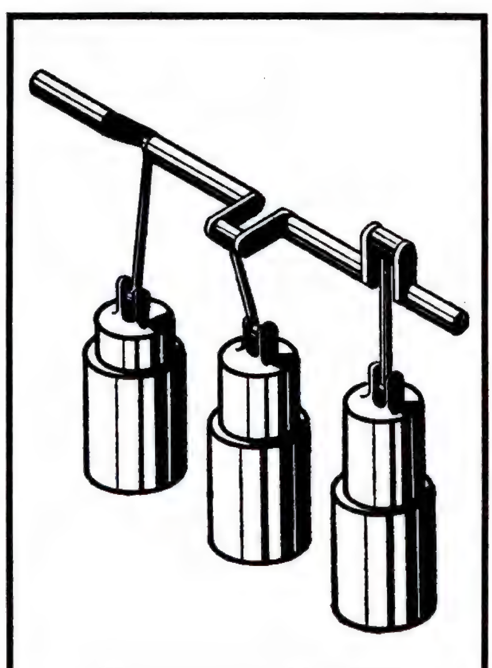

Figure 148

combined a certain distance from the pump, as likewise the two discharge pipes. With a double acting pump with combined discharge, flow is less pulsatory than with a single acting pump.

Varieties of set-up. Reciprocating pumps can be constructed to run horizontally or vertically. Vertical reciprocating pumps are often installed on bulkheads where they occupy space not otherwise usable.

Reciprocating pumps can also be constructed with more than one liquid cylinder driven by a single power source. Those with one cylinder are called simplex pumps, while those with two are called duplex and with three triplex. Still larger numbers of cylinders can be run by a single power source. The multiplication of cylinders increases the quantity of flow and decreases pulsation of flow. Triplex pumps are normally arranged so that each cylinder operates one-third of a cycle apart from the other two, as shown in Figure 148, so that no two are ever at the end of a stroke together. This arrangement reduces pulsatory flow to a minimum for this kind of pump.

Variable delivery reciprocating pumps. As already described, a

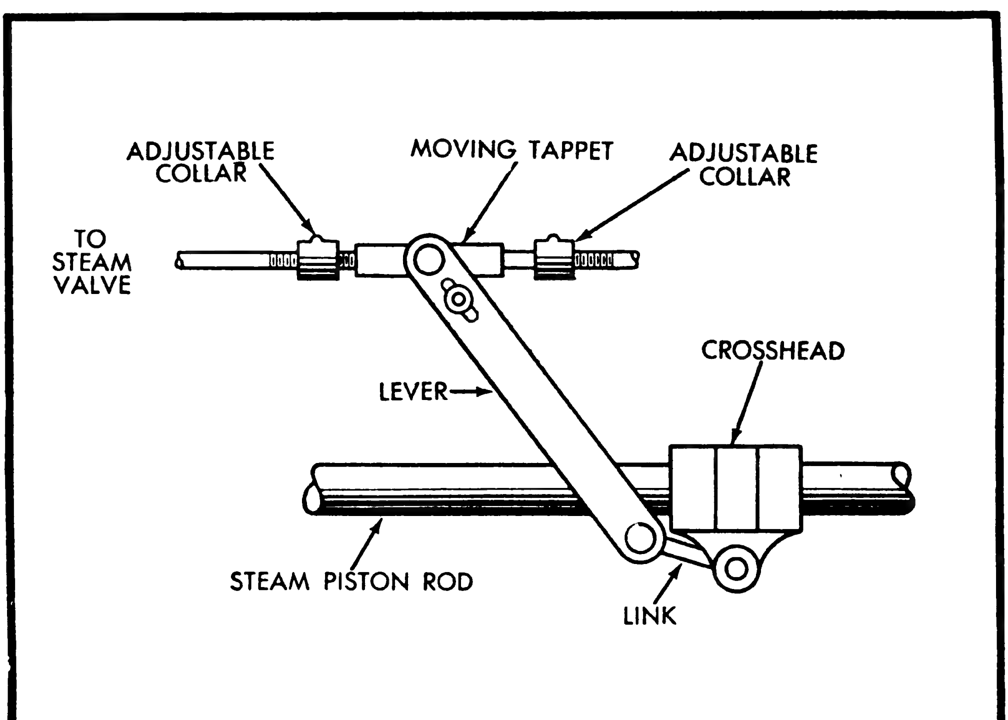

Figure 149

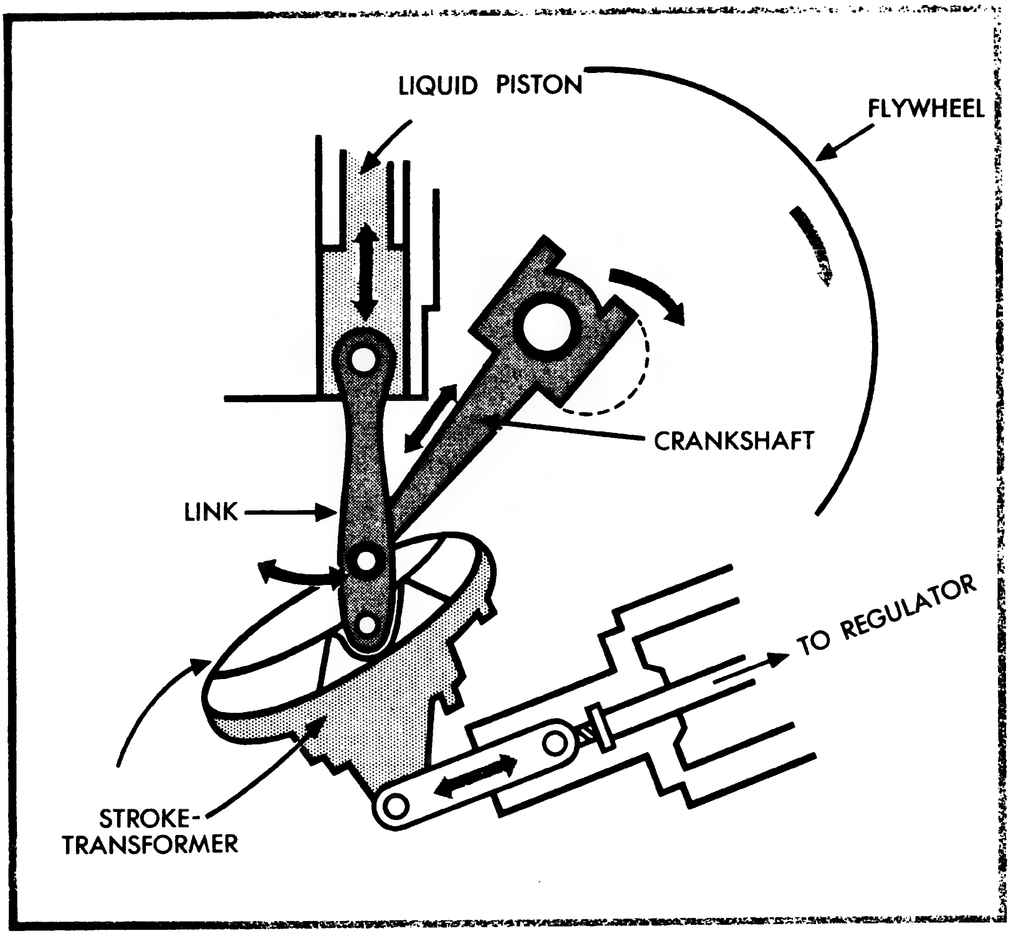

Figure 150

variable delivery pump is one in which the volume of delivery can be changed without altering the speed of the power source. To obtain variable delivery with a reciprocating pump the length of the piston stroke must be alterable. With direct acting pumps—those driven by steam with the two pistons in one straight line—variable delivery can easily be accomplished in the manner indicated in Figure 149. A crosshead moves back and forth with the steam piston rod. By a link unit and a lever, its movement is transmitted in reverse to a movable tappet which slides back and forth between two adjustable collars on a valve rod. The motion of this valve rod opens and closes passages for steam in the steam valve gear, so that the steam piston is driven back and forth. The distance between the adjustable collars determines the point at which the steam piston is reversed, and therefore, where the liquid piston changes its direction. If the adjustable collars are placed farther apart, the rate of delivery is increased.

In the Aldrich-Groff "Powr-Savr" power pump (Figure 150), the rotary motion of a turbine or motor is changed into a reciprocating motion by an ordinary crank shaft, and this in turn is transformed into a swinging motion which, finally, is communicated to a pump piston as a reciprocating stroke whose length varies with the angle of the swinging motion. This angle is determined by the set of a part called the "stroke-transformer." The set of the stroke-transformer can be automatically controlled by a hydraulic piston regulator.

Figure 150 diagrams the significant mechanical relations of this pump. A motor or turbine (not shown) rotates the fly wheel in a clockwise direction. This imparts a reciprocating motion to the crank-shaft, which is transmitted to the link as a swinging motion controlled by the angle at which the stroke-transformer is set. If the stroke-transformer was set in a horizontal position, the link would swing back and forth like a pendulum, and no up and down motion would be communicated to the hydraulic piston. In the position shown in Figure 150, the hydraulic piston would rise and fall a certain distance in its cylinder. For other angles of the stroke-transformer the length of its up and down motion would be different. The angle of the stroke-transformer is determined by the regulator (not shown in detail), which can be controlled by valves either worked by hand or automatically by the level of liquid in a reservoir, by pressure conditions elsewhere in the system, by rate of fuel supply, etc.

Navy uses of reciprocating pumps. Reciprocating pumps are used in the Navy for the following purposes:

a. Emergency feed.
b. Fuel oil booster and transfer (on older ships).
c. Fuel oil service (on older ships).
d. Air pumps.
e. Fire and bilge.
f. Evaporator feed.
g. Lubricating oil service (on older ships).
h. Miscellaneous intermittent services.

The air pump referred to in *d* above is a special type of reciprocating pump which is used to remove air and water from the condenser

of a boiler system. Its object is to maintain the vacuum in the condenser by removing air, and at the same time to draw off the condensate and deliver it to the feed tanks. Air pumps must be of large size, since they must draw a large volume of gas at low pressure, compress it to atmospheric pressure, and then discharge it. They have been largely replaced on vessels by air jet pumps of the kind described at the beginning of this chapter.

Maintenance and casualties. A full discussion of these topics is to be found in Chapter 47—*Instructions for the Operation and Maintenance of Pumps*—of the *Bureau of Ships Manual.*

Failure to start. In starting a steam driven reciprocating pump the throttle valve should be opened slowly so that the pump will warm up gradually. If the pump fails to start, the discharge line on the liquid side and the exhaust line on the steam side should be examined for closed or broken valves. The piston may have seized in the cylinder, especially if it has not been moved for some time. A bar may be used to apply pressure to move the piston, but care must be taken never to use a jacking bar to start a pump while the steam valve to the pump is open. If nothing is wrong with the valves or piston, it is probable that leakage of steam is taking place somewhere.

Faulty action. When a pump jerks on being started, it is failing to produce suction. A valve in the suction line may be closed, or the pump may be vapor bound and need to be cooled off. A pump that races without greatly increasing the discharge pressure may have a leaky piston, a leaky, broken, or stuck valve on the liquid end, or may be admitting air into the suction line. If the pump pounds, slow it down, install heavier springs in the suction valves, or charge the air chamber which is installed on most reciprocating pumps to regularize their action.

Making repairs. When repairs are undertaken, or parts replaced, a repair guide list should be used on which is stated exactly what was done or not done with respect to each condition found. This list should be preserved for reference at the time of the next overhaul. Section 47-51 of Chapter 47 of the *Bureau of Ships Manual* gives a

typical form which may be adapted to the requirements of each particular pump.

Chief trouble spots. If a pump works badly and is in need of repair, do not touch the power end until a thorough investigation shows that nothing is wrong with the liquid end. Most pump troubles are due to fouled liquid cylinder, worn valves, or conditions in the pipe connections outside the pump. Failure to keep pumps in proper alignment is one of the greatest causes of trouble aboard ship. Troubles in the steam cylinder result chiefly from leakage past the piston rings.

All pumps should be moved by hand daily, and under power once a week, so that parts will always move freely. If a pump runs on short stroke for some time, shoulders may be worn in the cylinder that will have to be removed before a full stroke will be possible.

QUESTIONS

1. What is a pump, and how do its main elements contribute to its action?

2. What different motions can be used to produce pump action? Illustrate each motion by giving a specific example of its use.

3. What are the five general types of pumps distinguished by this text? How does each type operate?

4. How are jet pumps used in Navy installations?

5. Describe the action of the check valves in a reciprocating pump.

6. Follow the path of liquid through an ordinary rotary gear pump.

7. What is centrifugal force?

8. Why is velocity head commonly transformed into pressure head in centrifugal pumps, and how is the transformation accomplished?

9. How does a propeller pump differ from a centrifugal pump in general structure?

10. What is the principal difference between positive and non-positive displacement pumps? Which pumps fall under each heading?

11. With which group of pumps is it dangerous to operate the pump with the discharge pipe closed? Why?

12. What is a variable delivery pump?

13. State the energy relations for a pump standing above the level of the liquid being pumped.

14. What is the relation of suction head to atmospheric pressure?

15. What is meant by the characteristics of a pump?

16. On what grounds can pumps of different types be compared? What are the general advantages of each type?

17. What means can be employed to avoid air in a hydraulic system?

18. What are some of the signs of trouble in a pump?

19. What is a direct action pump?

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Chapter 9

CENTRIFUGAL AND PROPELLER PUMPS

This chapter describes two types of pumps—the centrifugal and propeller types—which differ in the motions utilized, and then proceeds to consider their combination in mixed flow pumps. All three kinds of pumps are made in a variety of designs for one or another specific purpose. Some of these variations will be described. All of the pumps considered in this chapter belong to the non-positive or continuous flow group, as distinguished from reciprocating and rotary pumps, which are of the positive displacement variety.

What Goes On in a Centrifugal Pump

As was briefly pointed out in Chapter 8, centrifugal pumps make use of rotary motion, and involve the joint use of two separate actions. We must now explain these two actions and show how they are combined in the operation of the centrifugal pump.

No new principles beyond those already stated in Chapters 1 and 2 will be necessary for the understanding of centrifugal pumps. We shall approach them by way of the principle of inertia, which was defined, explained, and illustrated in Chapter 2. For convenience we shall repeat the basic law of inertia here: "A body at rest tends to remain at rest, and a body in motion tends to continue in motion

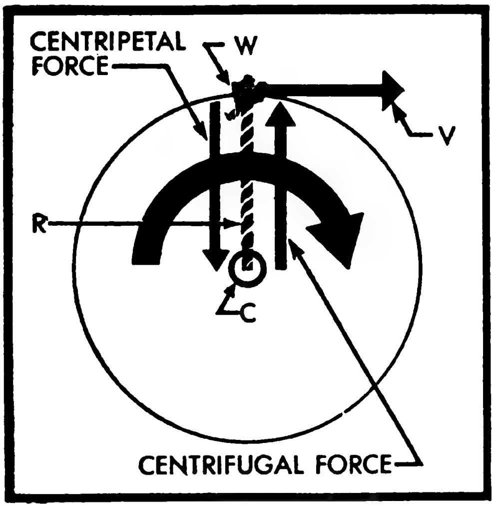

Figure 151

with the same velocity and in the same direction."

The action of centrifugal force. In the analysis and illustrations given in Chapter 2, attention was confined to straight line motion and to the relationship of inertia to forces tending to speed up or slow down an object. However, as will be noted from the basic law, the question of the direction of motion must also be considered. Since an object in motion tends to continue to move in the direction in which it is going, a force must be applied to it acting at an angle to its direction of motion, if we wish to make it move along any path other than a straight line. The object will continue to change direction so long as the force acts at an angle to its direction of motion. If the force is removed, the object will immediately travel in a straight line in the direction in which it was moving at the moment that the force ceased to act.

In Figure 151 we have an object *W* which is connected by a cord to a fixed point or center *C*, around which it is being swung. At the instant under consideration, the object has a velocity *V* in the direction shown. According to the law of inertia the object would continue to travel along the straight line *V* at the same velocity unless some force interferes. It will readily be seen that in order to travel along *V* it would have to stretch the cord *R*. If this cord is strong enough to resist being broken or materially stretched, it will exert a force on the object to change its direction of motion, but not its velocity. Since *R* is a fixed length, the object will travel in a circle of radius *R* about the center *C*. In so doing its direction of motion will be continually changing, which means that the cord will be continually pulling upon it.

This pull which the cord exerts on the object is called centripetal force, and its reaction, the force which the object exerts on the

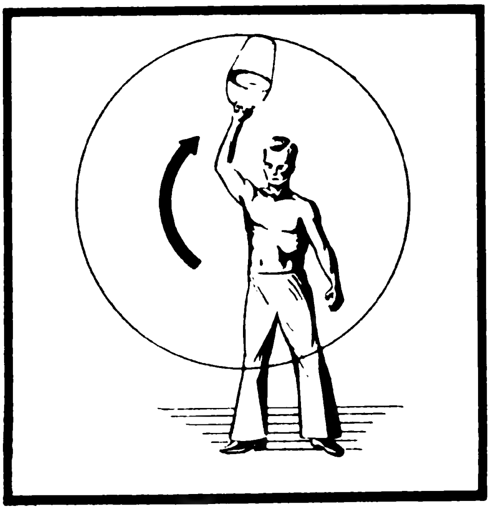

Figure 152

cord, is called centrifugal force. These forces always act along the radius of the circle, and therefore at right angles to the direction of motion of the object at the moment under consideration. For this reason the velocity of the object does not change, and if friction were ignored the object would continue to swing around the center C, putting a continuous tension on the cord. The magnitude of this tension would depend upon the mass of the object, its velocity, and the length of the cord R.

Many illustrations of centrifugal force could be cited. A familiar one is the tendency of an automobile to skid towards the outside of the curve when rounding a corner at high speed. For the same reason the occupants of the car are forced against its outer side. A simple demonstration of centrifugal force can be arranged by whirling the water in a pail or basin. The water level will rise at the edges and descend in the center in a miniature whirlpool effect. The difference in level is caused by the centrifugal force carrying the water away from the center towards the circumference.

Centrifugal force in a pump. An example of centrifugal force helpful for the understanding of centrifugal pumps can be found if one imagines a man whirling a bucket of water rapidly around and around in a circle, as shown in Figure 152. If he whirls the bucket rapidly enough, the outward or centrifugal force will hold the water against the bottom of the bucket and it will not spill, even when the bucket is upside down. By whirling the bucket very rapidly, a force many times the force of gravity can be developed.

To understand the application of this illustration to a centrifugal pump, let us assume that we have a number of bottomless buckets rotating about a center, as shown in Figure 153. The bottom of the buckets are sealed by contact with what might be called a con-

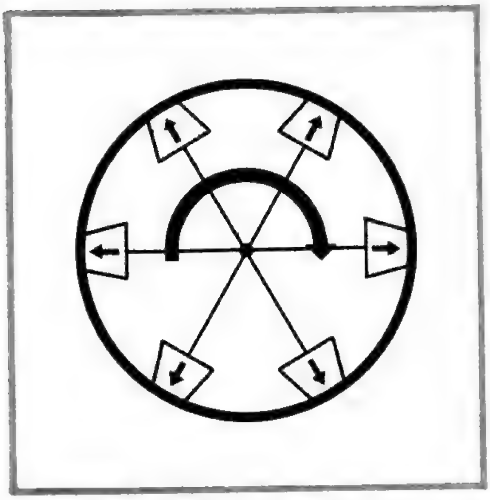

Figure 153

tinuous bottom in the shape of the bounding wall against which they rotate. As the buckets rotate, centrifugal force pushes the liquid against this continuous bottom.

Now let us imagine that the buckets are shaped like pieces of pie, and that they completely fill the circle, as shown in Figure 154. We have in effect a paddle wheel revolving in a drum-shaped container. The result will be a continuous liquid pressure due to centrifugal force all over the outer circumference of the container.

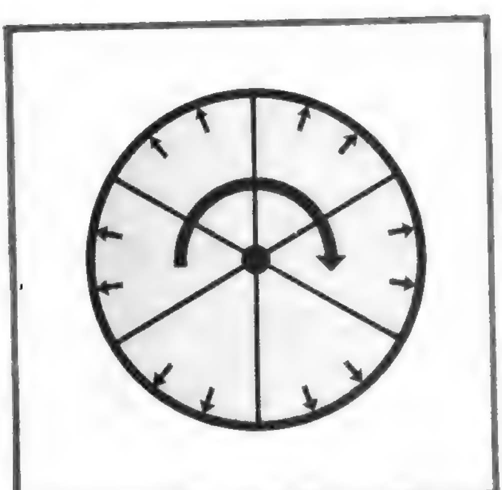

Figure 154

Now let us enlarge the diameter of the container so as to leave a space *A* for liquid between the ends of the paddles and the drum,

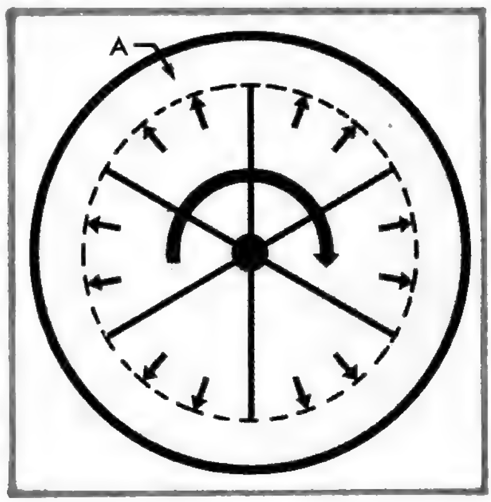

Figure 155

as shown in Figure 155. We now have a liquid bottom for our wedge-shaped buckets, but the action is just the same as before. The rotation of the liquid in the buckets, because of the centrifugal force, pushes outward against the liquid bottom and thereby imparts a pressure to the liquid in space *A*. If we assume that this liquid remains stationary—which would not actually be the case in practice—then it would be subjected to a true

static pressure the magnitude of which would be dependent upon the centrifugal force of the liquid between the paddles.

If an opening were provided in the containing drum, this pressure would cause the liquid in the space *A* to be discharged. If the liquid in space *A* were stationary, this discharge could take any direction at all because, according to Pascal's Law, static pressure in a liquid is transmitted equally in all directions. In practice, however, another factor must be considered which determines the direction of discharge, as will be explained shortly.

If now we add a means for continuously filling the buckets, as by a central opening in the container, and a source of power to rotate the paddles, we will have a continuous flow, as shown in Figure 156, demonstrating one of the two basic principles utilized in the centrifugal pump.

Tangential action. In our general discussion of the subject of centrifugal force it was pointed out that, the instant the force which is compelling the object to move in a curved path is removed, the object will travel in a straight line in the direction and with the velocity it had at the instant this happened. This is illustrated in Figure 157, where it is assumed that the cord *R* is broken when it reaches the vertical position shown. The object will then fly off horizontally (at right angles to the cord) with the velocity it had at that moment. This horizontal path will be tangent to the circular path the object was following up to this moment, and so this action is commonly called "going off on a tangent", while the velocity is called tangential velocity.

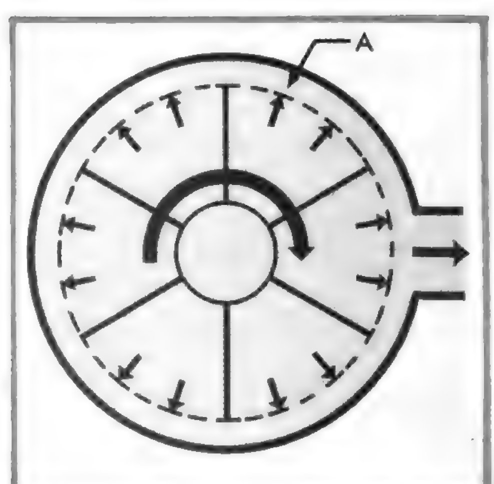

Figure 156

This is the principle of the old fashioned sling shot, in which a man placed a stone in a leather holder to which two strings were

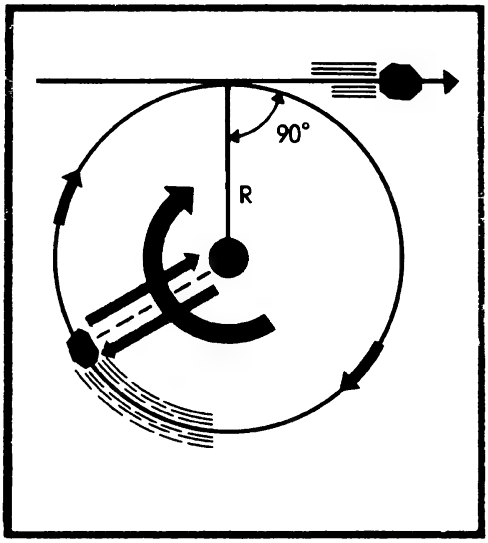

Figure 157

attached, swung it around his head to impart velocity, and then let go one of the strings. This principle is utilized in the centrifugal pump in addition to the outward action of centrifugal force already described.

Joint result of the two actions. Returning again to Figure 156, it will be clear that as soon as some of the liquid in space *A* is permitted to escape, a continuous low will be set up. Liquid will enter the container through the center opening or eye. The blades or paddles will then start rotating it. As the liquid in *A* flows out of the container, the liquid between the blades will move out to a greater and greater radius. In so doing it will acquire an increasing tangential velocity because the angular velocity of the blades is constant and as the radius of the path circle increases, the liquid must travel a greater distance per revolution. Eventually it will reach the tip of the blades and enter space *A*, which is equivalent to "going off on a tangent" as just described. The liquid will also tend to continue moving directly outward from the center of the circle because of its inertia, as already explained. In other words the two actions—centrifugal and tangential—will be combined.

The liquid which is thrown off the tips of the blades, as already pointed out, has a velocity equal to the tangential velocity it had at the instant it left them. Therefore it has a velocity head, and if the discharge opening is increased in size this velocity head will be converted into pressure head in the space *A*. Therefore, we have two separate actions operating simultaneously to produce pressure in the space *A*. One is the strictly centrifugal action, which acts radially, and the other is the tangential action, operating along a tangent which is always at right angles to the radius at each particular instant.

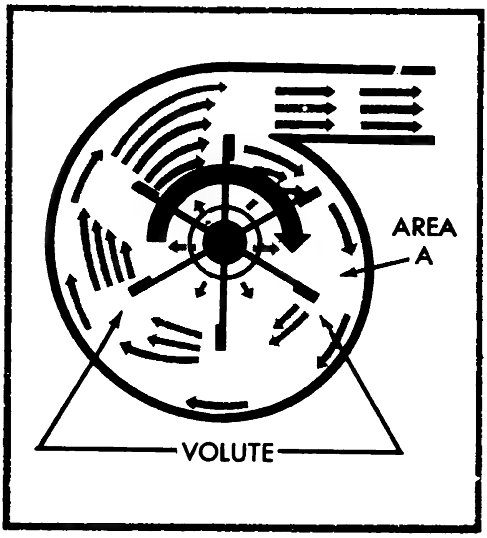

Figure 158

Design considerations. Since it is part of the meaning of velocity that it have a direction, the direction in which the discharge takes place becomes important. In order to make the most use of the velocity head, the discharge is generally in an approximately tangential direction, but the exact size and shape of the chamber and the precise direction of discharge are matters which concern only the designer of centrifugal pumps.

It will be noted that liquid is being thrown off the paddles, or impeller blades as they are called in the pump, all around the circle. Therefore liquid is being added to the space *A* at all points, but is escaping from it at only one point. In order to compensate for this, it is customary to increase the area of the space *A* as the outlet is approached. By this means it is also possible to slow down the tangential velocity at a more gradual rate, and thereby obtain more efficient conversion of the velocity head into pressure head. Therefore, it is customary to give the space *A* a shape like that shown in Figure 158. This shape—called a *volute*—continuously expands in a definite ratio. The details of the size and shape of the volute used are very important to the efficiency of the pump, as are likewise the exact shape, size and speed of rotation of the impeller. In general it is true that impeller blades that curve backwards with respect to their direction of rotation give better results than straight blades, but the reasons for this and for other details of construction are matters which concern the designer primarily and are outside the scope of this book.

Kinds of Centrifugal Pumps

Centrifugal pumps come in three designs—volute, diffuser and peripheral turbine. Since the first two are quite similar, they will be considered together.

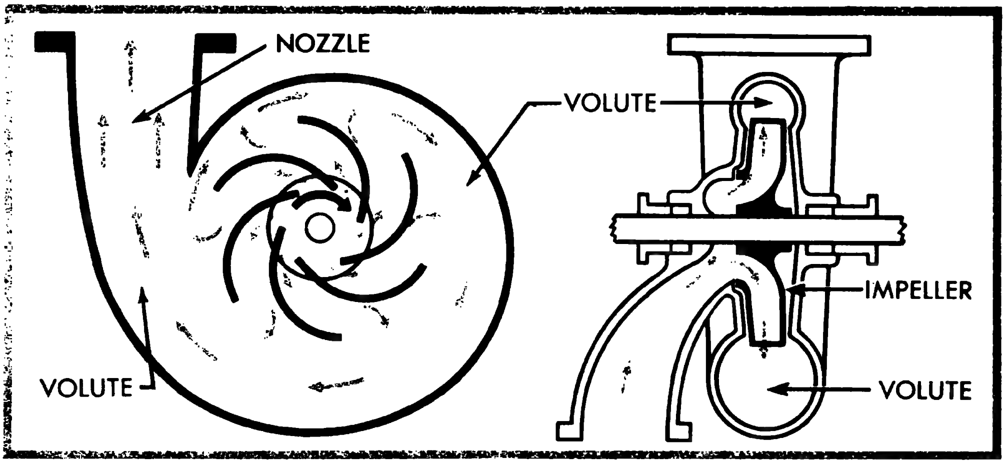

Figure 159

Volute and diffuser centrifugal pumps. In these pumps pressure is developed by the action of centrifugal force pushing the liquid away from the hub of the impeller, while at the same time the rotation of the impeller produces the greater part of the velocity which drives the liquid out through the discharge.

In the volute pump—the one most commonly used—the impeller discharges into a progressively expanding casing, as shown in Figure 159. The casing is proportioned to produce equal velocity of flow all around the circumference of the casing and then gradually to reduce the velocity as the liquid passes from the casing into the nozzle to be discharged from the pump, thereby transforming a considerable part of the velocity head into pressure head.

In the diffuser pump, the impeller is surrounded by gradually expanding passages formed by stationary guide vanes, as shown in Figure 160. In these expanding passages the direction of flow is changed and velocity head largely converted to pressure head before the liquid enters the volute. It is worth noting that the

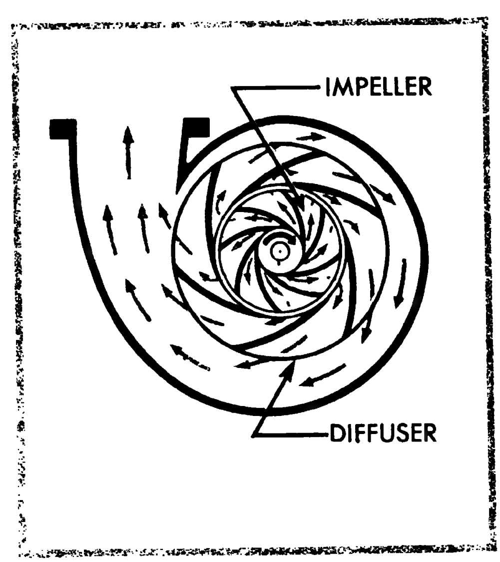

Figure 160

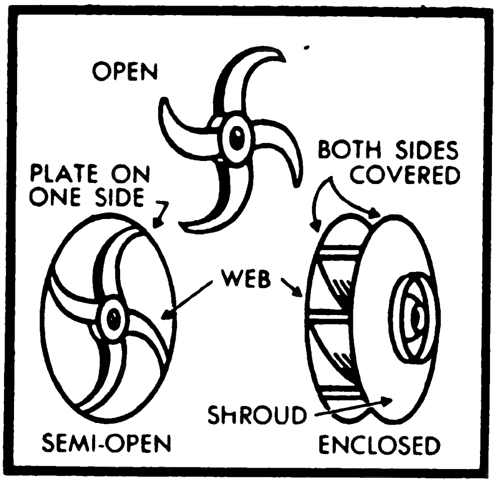

Figure 161

diffuser vanes are set approximately tangent to the ends of the impeller blades. In this kind of casing, efficiency may be slightly higher than in volute pumps, since velocity head is more completely converted into pressure head. The added cost of manufacture and more complicated construction of diffuser pumps, however, is generally not considered justified except occasionally in the case of large high-pressure pumps.

The impellers used in volute and diffuser pumps are of three kinds —open, semi-open and enclosed, as shown in Figure 161. The open impeller consists only of blades attached to a hub. The semi-open impeller is constructed with a circular plate — the *web* — on the inside edge of the blades. The web need not extend all the way out to the ends of the blades. The use of a web makes it possible to use thinner blades. In the case of the enclosed impeller, a *shroud* is added on the outside edge of the blades, so that the liquid is in large measure confined in the blade region, between the web and the shroud. Holding the liquid between these plates reduces friction losses in the pump.

Impellers may also be single or double suction (Figure 162). The former consists of a single impeller drawing liquid in but one direction, while the double suction impeller is similar to a pair of single suction impellers placed back to back, so that the liquid is drawn into the pump from opposite directions.

When liquid enters a centrifugal pump a certain amount of *end thrust* is produced, due to the unequal forces acting in opposed directions along the axis of rotation. Thus in the single suction pump of Figure 162, the outside faces of both the web and the shroud are subject to the discharge pressure, but the area of the web is greater than the area of the shroud by an amount equal to the size of the

eye. A force acting back towards the place of entrance of the liquid is therefore left unbalanced, although it is somewhat diminished by the fact that a practically equal area on the inside face of the web is exposed to suction pressure, which is of course very much less than discharge pressure. There are other causes of end thrust, but they all result in a hydraulical force acting towards suction which must somehow be balanced. This can be done by bearings, by directing flow in the pump to each side of the web and shroud so as to improve its hydraulic balance, or by use of a double suction set-up, where end thrust in one direction is balanced by end thrust in the other. This principle is easily seen in the double suction diagram of Figure 162.

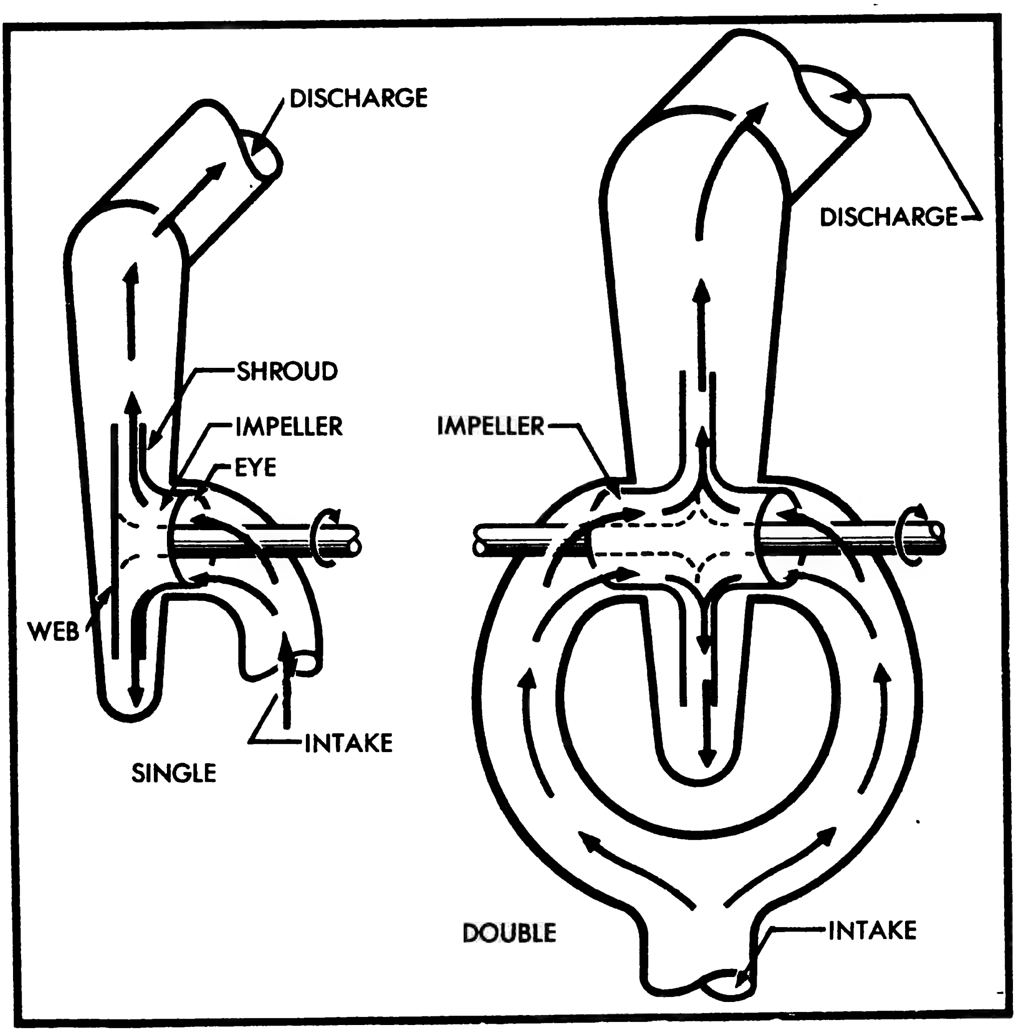

Figure 162

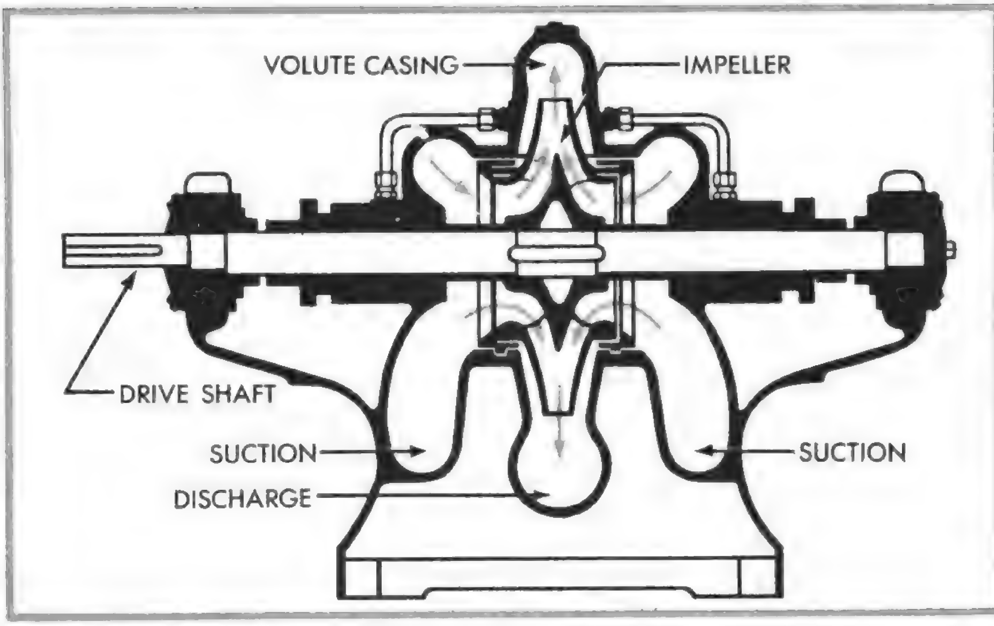

Figure 163

Single and multistage pumps. Single stage pumps contain only one impeller, or one double suction impeller group, drawing in the liquid and expelling it through the discharge into the hydraulic system. A double suction single stage pump is shown in Figure 163. A disadvantage of the single stage pump, however, is that pressure cannot be increased above the working maximum of the impeller. Above this limit, under practical working conditions, the pump will not deliver any further increase in pressure.

In multistage pumps, this disadvantage is overcome by combining several single stage pumps, whether single or double suction, so that the discharge of one impeller is delivered to the suction of the next impeller. The liquid is delivered to each succeeding stage under the pressure imparted to it by the preceding stage, and additional velocity or pressure is added. As the liquid passes through each impeller in turn, additional pressure is imparted to it. A much higher working head can be produced than is possible with a single impeller. For the sake of compactness the several impellers of multistage pumps are almost invariably placed on one shaft, and the whole unit is built into one housing. The impellers are arranged in multistage pumps to eliminate or minimize end thrust. Figure 164 shows a possible flow diagram for a four-stage pump.

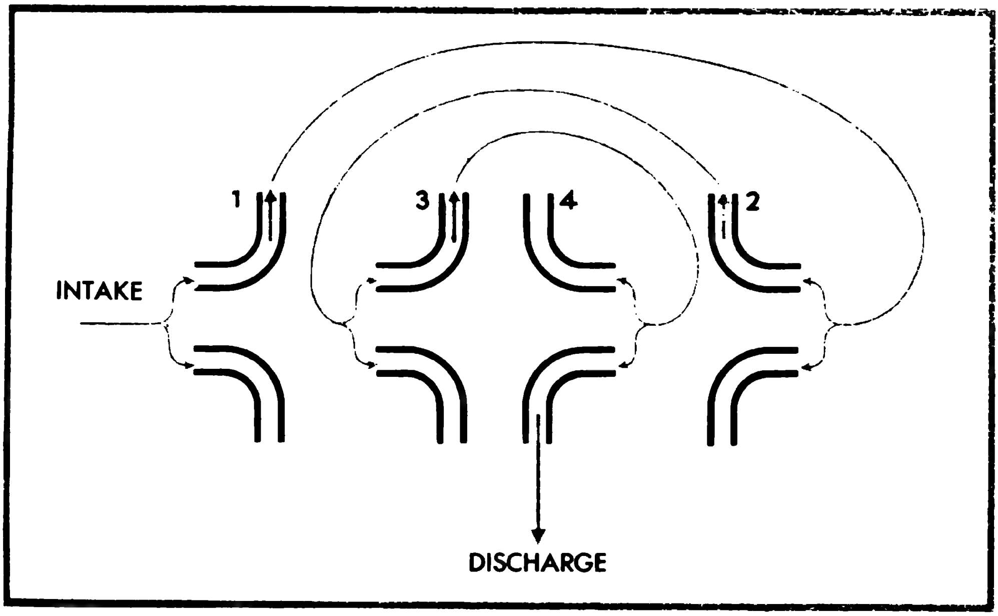

Figure 164

Both volute and diffuser pumps will be found in single and multistage construction, although diffuser pumps are almost invariably multistage. Because of the greater cost and complexity of diffuser pumps, they are not often found aboard naval craft.

Our illustrations have emphasized horizontal pumps—pumps, that is, with their axis set in a horizontal plane. These pumps can operate just as efficiently when their axis is set vertically. A vertical centrifugal pump is shown in Figure 165.

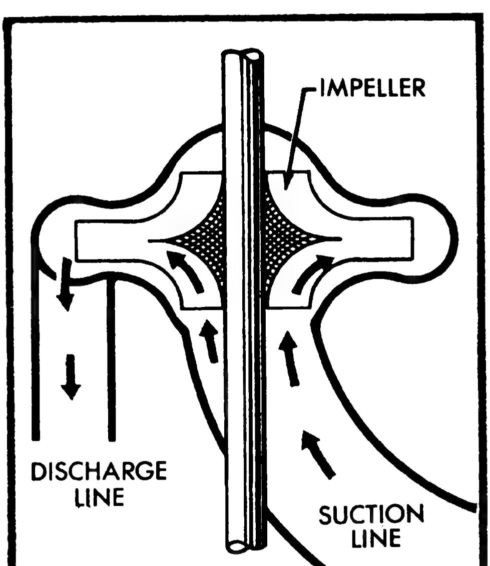

Figure 165

Because of their simplicity, low cost, and ability to operate under a wide variety of conditions, centrifugal pumps are widely used. They are adapted to operate under practically any head up to several thousand feet, and will handle liquids at temperatures up to 1000° F, as well as liquid containing a high proportion of

abrasives or even rather large solid particles. Just as they have tended to displace reciprocating pumps for some purposes, however, they in turn have been displaced to some degree by the development of rotary pumps, especially where precise control and variation of delivery are required during the course of a single work operation.

The relationship between volume of flow and pressure developed by a pump is expressed in a curve called the *characteristic* of the pump (see Chapter 8). For each centrifugal pump this curve has a top limit, and therefore the discharge of the pump can be closed and pressure in the pump will not rise higher than this value, although the pump continues to run. The designer of a centrifugal pump, however, can make the curve for the pump he is constructing almost anything he wishes by changing different features of his design.

Casualties and their remedies. Centrifugal pumps will give good service if given reasonable care and if operated under the conditions for which the pumps were designed. When troubles do arise, however, the following suggestions should be helpful in locating and correcting them.

1. Pump fails to start pumping.
   a. Pump is probably not properly primed. Reprime the pump, opening all the vents until a steady and unbroken stream of liquid is secured.
   b. Impeller may be clogged. Examine carefully for solids or foreign matter lodged in the impeller.
   c. Strainer or suction line may be clogged. This would cause an excessive lift.
   d. Wrong direction of rotation. Arrow on pump case shows the proper direction of rotation.
   e. Speed may be too low. Check the speed with speed counter or tachometer and compare with speed called for on name plate of the pump. When the pump is being driven by electric motor, check up to see whether the voltage may not be low. When driven by steam turbine, make sure that the turbine is receiving full steam pressure.

f. Suction lift too high. Check with vacuum gauge or by actual measurement where possible, adding to the static suction lift the friction due to the pipe and fittings.

g. Total static head is probably much higher than that for which the pump is designed. Check with dependable vacuum and pressure gauges; or preferably, determine by actual measurement the difference in elevation between pump and point of discharge. Add to this static head the friction due to the pipe and fittings used in the installation.

2. Outgo not up to capacity and pressure.

a. Air may be leaking into the pump through suction line or through stuffing boxes.
b. Speed may be too low.
c. Discharge head may be higher than anticipated.
d. Suction lift may be too high.
e. Foot valve or end of suction line may not have sufficient submergence, or it may be partly clogged or entirely too small.
f. Impeller may be partly clogged or too small in diameter.
g. Rotation may be in the wrong direction.
h. Wearing rings may be worn, impeller may be damaged, shaft sleeves may be loose, stuffing-box packing may be defective, or casing gasket may be torn.
i. If hot or volatile liquids are being pumped, there may not be sufficient positive head on the suction.

3. Pump starts and then stops pumping.

a. Improperly primed pump or leaky suction line.
b. Air pockets in suction line.
c. Water-seal line may be plugged.
d. Air may be entering suction pipe because liquid is being delivered in the suction tank or sump too near the pump suction line.
e. Suction lift may be too high.

4. Pump takes too much power.

a. Speed may be too high.
b. Head may be too low and pump delivers too much liquid.

c. Liquid may have too high a specific gravity.
d. Pump may be operating in wrong direction.
e. Casing may be distorted owing to excessive strain from suction and discharge piping; shaft may be bent; impeller may be rubbing on casing; rotating element may be binding; stuffing boxes may be too light; wearing rings may be worn; casing packing may be defective.

5. Miscellaneous.

a. If coupling bushings wear too rapidly, check up the alignment.
b. If bearings run hot, they are probably overfilled with lubricant if of the antifriction type; insufficient lubricant if bearings are of sleeve type.
c. If there is any undue vibration of the pump, the impeller is probably partially clogged.
d. If the motor is running hot, the voltage is probably low or else the speed is too high.

Peripheral turbine pump. This pump is often called a turbine or

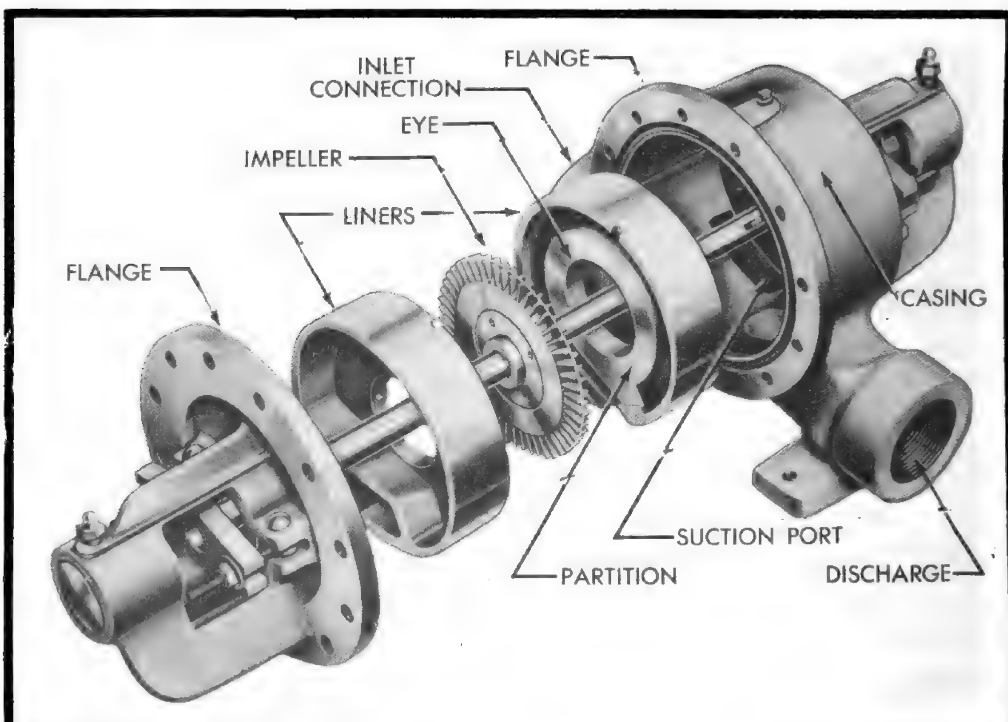

Figure 166

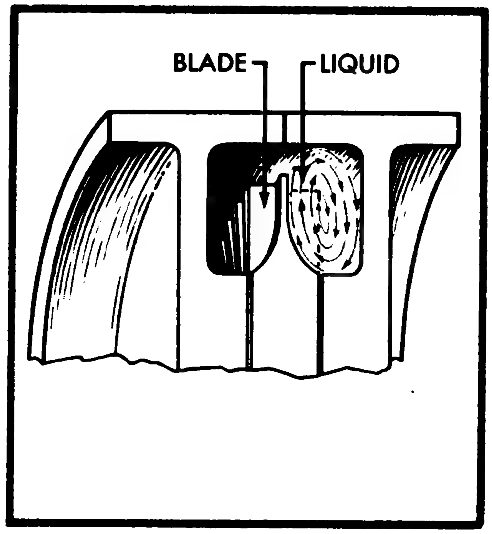

Figure 167

regenerative pump. While it differs markedly from volute and diffuser pumps, it does belong in this group. Figure 166 shows an exploded view of a Westco turbine pump. In this view, the impeller and the two liners fit into the pump casing, the flange faces at the left and the right meeting and being bolted together. The inlet or suction connection is at the back of the pump casing. It divides into two suction ports (one not shown), which deliver liquid to both sides of the impeller through eyes in the liners. The action of the pump carries the liquid out to the impeller blades. After the liquid has made practically a complete revolution around the pump chamber in the channel which houses the impeller blades between the two liners, it is diverted into the discharge, which is separated from the suction by a partition. In other turbine pumps, liquid is delivered directly to the blades at the periphery of the chamber.

In the other pumps discussed in this chapter, liquid passes between the impeller blades but once, and receives energy only while travelling the short distance along the blade from the center to the outside edge of the impeller. In the peripheral pump, the liquid circulates from blade to blade as it passes around the circumference of the pump, and is whirled around by the blades as shown in Figure 167. The energy supplied to the liquid is a summation of the impulses it receives from a large number of impeller blades striking it one after the other as it is carried around the outer edge of the pump chamber.

For high suction lift the peripheral pump is ideal. It has the highest practical lift characteristic of any of the liquid-handling pumps, as shown in Figure 168, which makes it an excellent unit for handling hot and volatile liquids. For suction lift the turbine pump is

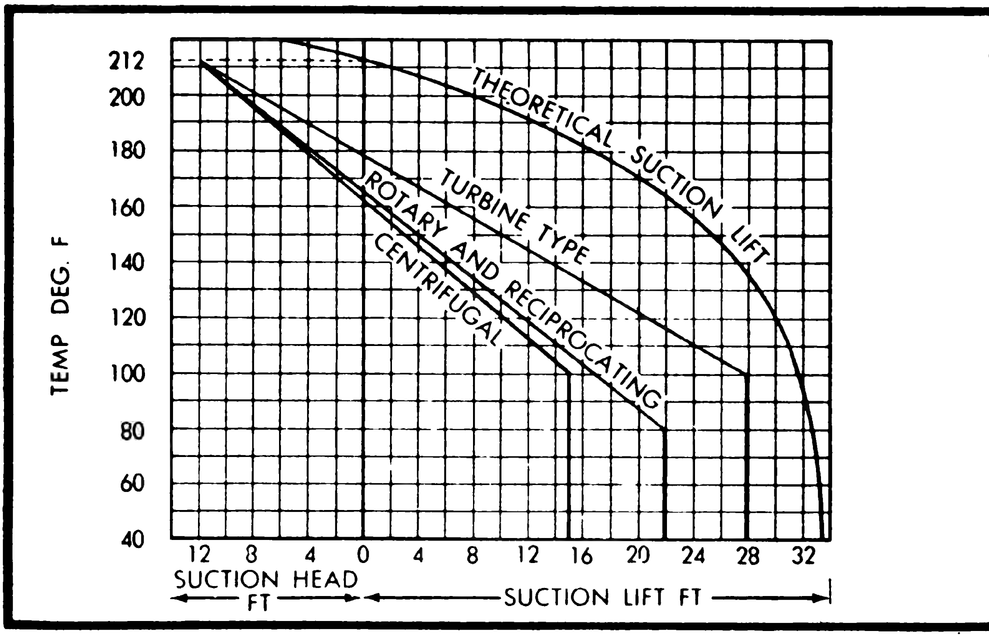

Figure 168

matched only by some of the more modern designs of rotary pumps. Because of the rather close clearances of turbine pumps, they are not suited for handling highly viscous liquids, or those carrying abrasives.

The peripheral turbine pump is usually a single stage pump, since it is possible to develop a high head in but one stage. The impeller blades are made in several designs, including flat vanes, V-shaped vanes, and curved buckets.

Propeller Pumps

Pumps of this kind are similar in action to the propeller of a boat encased in a tube. Liquid is drawn into the pump along the axis of rotation, and is pushed out without changing its direction of flow, as shown in Figure 169. Propeller pumps have very little suction or lifting power, but they do have a fairly high pushing power that can be applied to large volumes of liquid.

Propeller pumps must be located below or at no great distance above the surface of the liquid to be pumped. They are usually employed where the discharge head is under 40 feet and volume of flow is

above 300 gallons per minute. They find wide use in drainage, sewage, storm water disposal, irrigation, and condenser circulating water systems. Propeller pumps are marketed under a wide variety of names, among which are axial flow, straight flow, screw type, spiral type. All such pumps employ a common principle of operation.

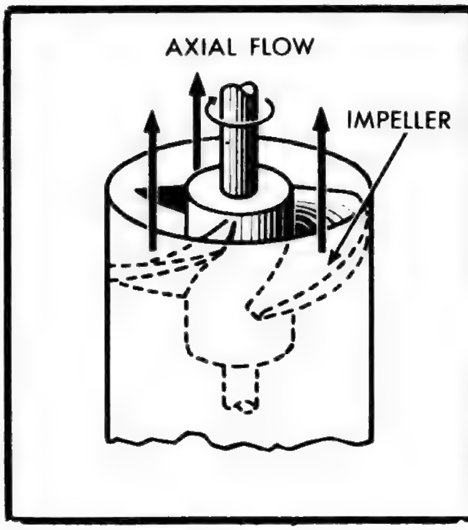

Figure 169

Figure 170 shows a typical propeller from one of these pumps. The propellers are encased in a tube of slightly larger diameter than the propeller. The tube usually has vanes incorporated in it to counteract the swirling motion given to the liquid by the rotating blades. Figures 171 and 172 show two varieties of this pump. Propeller pumps are as a rule single stage, although they are sometimes constructed in multi-stage units (Figure 173).

Propeller pumps may be built to stand either vertically or horizontally. The vertical types are very compact and can be placed in the suction well so that the pump will always be primed. The actual horizontal area required for the pump need be no greater than

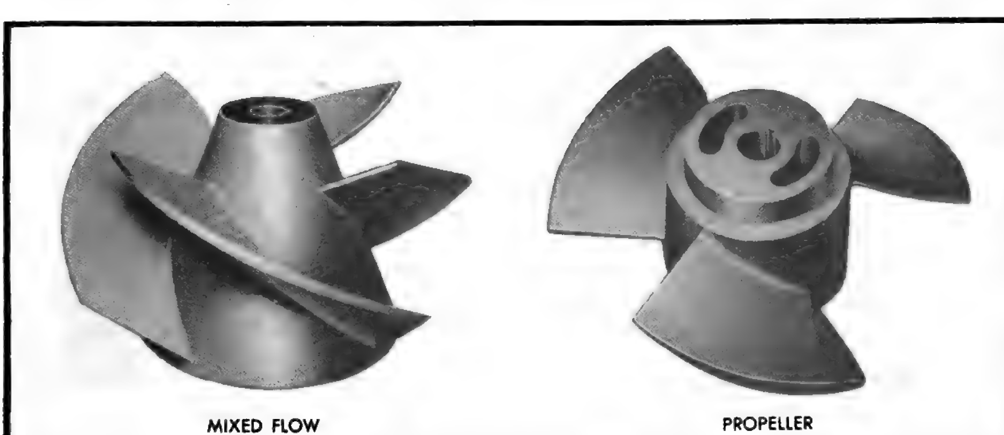

Figure 170

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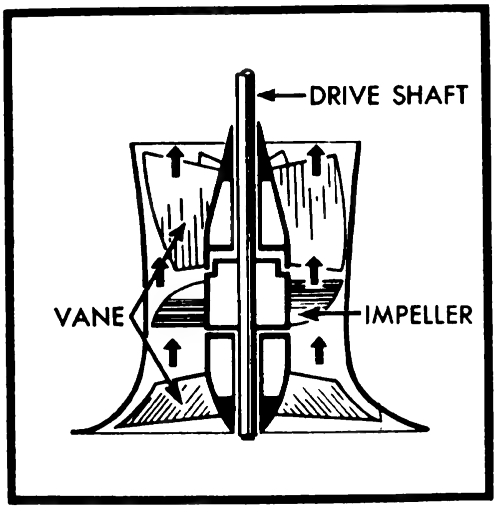

Figure 171

is needed for a vertical motor. Horizontal propeller pumps are also very compact, often being not much larger than the pipe which carries the liquid. With both varieties—vertical or horizontal—provision must be made

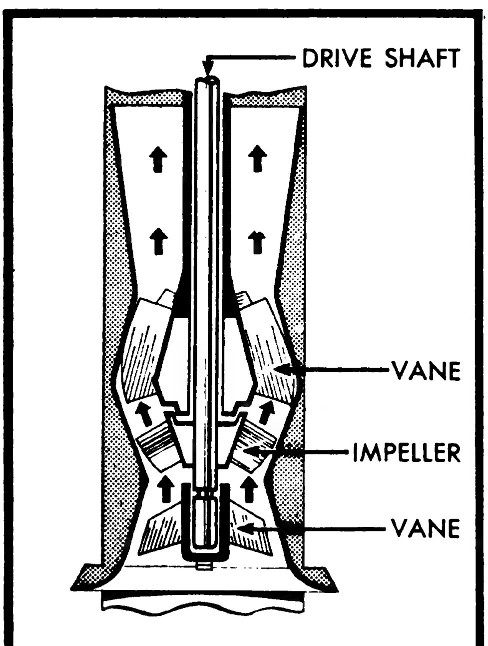

Figure 172

for priming when the pump is located above the liquid. Priming should always be done from the highest point in the pump casing to make certain that no air pockets remain in the pump.

Maximum efficiency for the ordinary propeller pump is maintained only over a rather narrow range of discharge and pres-

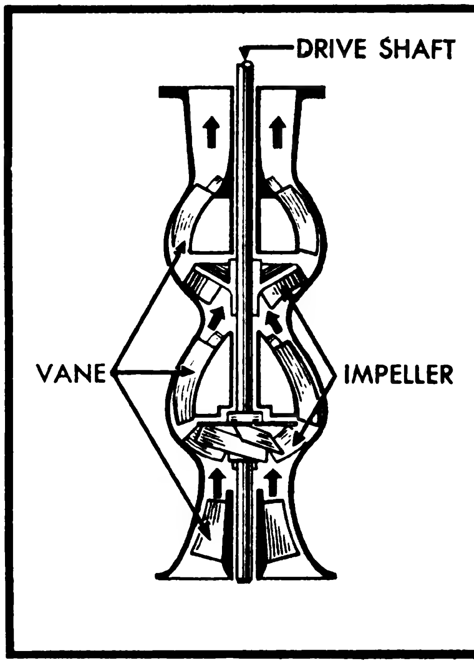

Figure 173

sure produced. For large installations, as in the Tennessee Valley projects, this would be a great disadvantage, since pumps under such conditions must deal with wide variations of input and output. The disadvantage was overcome by the development of a propeller pump with automatically or manually adjustable

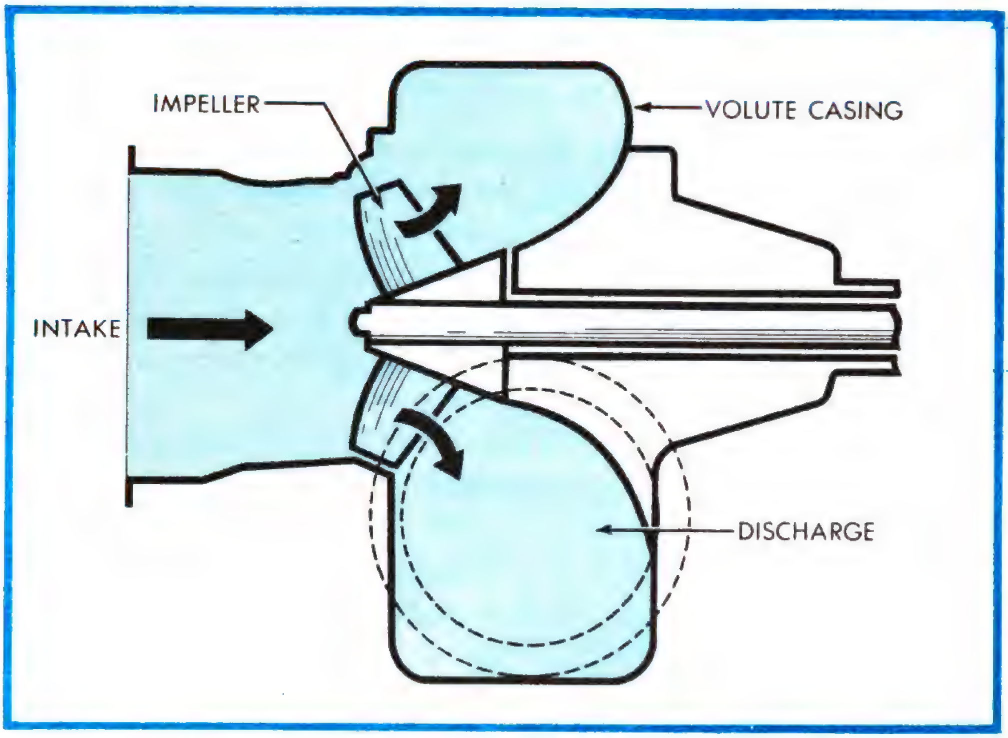

Figure 174

blades—the so-called Kaplan turbine. There is no occasion to use such pumps on board ship, however.

Mixed Flow Pumps

A mixed flow pump is one in which the head is developed partly by centrifugal force and partly by propeller action. A typical propeller is shown in Figure 170. This pump as shown in Figure 174 always has a single inlet impeller with the flow entering along the axis of rotation and discharging both axially and radially in other words, both as it would discharge from a propeller pump and from a centrifugal pump. Mixed flow pumps have volute casings like most centrifugal pumps, and may be constructed

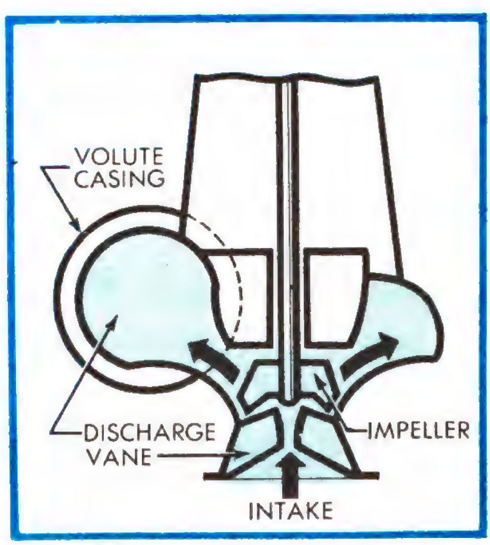

Figure 175

to work either horizontally as shown in Figure 174 or vertically as shown in Figure 175.

Mixed flow pumps must be operated at fairly high speeds to be effective and economical. They can handle liquids that are very viscous or that contain dirty or gritty materials more easily than either the centrifugal or propeller types, since the impeller blades can be spaced farther apart. The centrifugal force imparted to the liquid makes it possible to obtain more head from this pump than from the straight propeller type.

QUESTIONS

1. What is centrifugal force?
2. What is a tangent to a circle?
3. What is a volute curve?
4. What are the three kinds of centrifugal pumps?
5. What is the difference between volute and diffuser pumps?
6. What kinds of impellers are used in centrifugal or diffuser pumps?
7. What is the difference between single stage and multistage pumps?
8. How does the peripheral turbine pump differ from volute or diffuser pumps?
9. How does a liquid receive its energy or head in a centrifugal pump?
10. What kinds of impellers are used with the peripheral turbine pump?

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Chapter 10

CONSTANT DELIVERY ROTARY PUMPS

This chapter describes constant delivery rotary pumps, first in a general way with respect to the manner in which they operate. The different varieties that have been developed and their fields of use are then considered, after which some typical pumps of this kind are described in more detail, and the problems connected with their operation are discussed.

General

History. The principle of action of rotary pumps has long been known. There is record, for example, of an excellent French gear-type rotary pump dating from the early seventeenth century. The rotary pump was not used extensively, however, until late in the nineteenth century when the electric motor came into general use. This source of power is well adapted to drive rotary pumps, since it is only necessary to connect two rotor shafts.

At first rotary pumps were relatively inefficient, except for situations requiring small volumes at large heads or large volumes at small heads. Today, the number of uses to which these pumps have been put has rapidly increased, and as time goes on their field of application continues to expand. This is especially the case since varieties of the rotary pump have been developed which can be adjusted to give variable delivery without change in the speed of

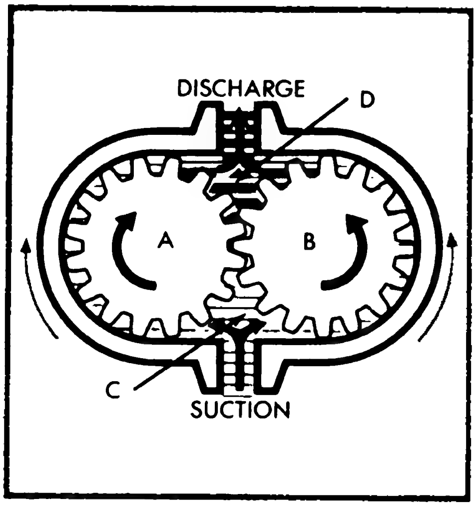


Figure 176

operation of the source of power. These pumps will be described in Chapters 11 and 12.

Operation. All the many varieties of rotary pumps operate on the same principle. An element called the *rotor* is rotated in the pump chamber in such a way that liquid is carried or pushed from the suction side of the pump to the discharge side. Figure 176 illustrates this action with a gear pump. The liquid is carried from suction to discharge in the spaces between the gear teeth and the surface of the pump casing as the gears rotate. One of the gears is directly driven by the source of power, while the other rotates with it, in the opposite direction. This is accomplished either because motion is imparted from the drive gear to the idler gear by the meshing of the two gears at the center of the pump chamber, or because *timing gears* standing outside the pump chamber transmit motion from one gear to the other. Figure 188 shows timing gears in a variety of pump to be described later.

There are close clearances between the gear teeth and the pump casing, and between the teeth of the two gears at their point of contact, where they form a continuous fluid-tight joint. As the gears rotate in the direction indicated by the arrows, liquid is trapped in turn between each pair of teeth and the casing, and carried away from the intake chamber. At the same time, as teeth unmesh at the center, the space they occupied is empty of liquid. Pressure is therefore lowered in the intake chamber, so that liquid flows into it from the source of supply as the gears rotate.

Rotary pumps are of necessity positive displacement, since they deliver a definite quantity of liquid for each revolution of movement. They tend to continue delivering liquid if the discharge is closed or blocked. If the discharge of a rotary pump were closed

with the pump operating, and if no unloading valve of any kind were in the circuit, pressure would build up in the pump until the pump packing would fail, the driving motor overload mechanism would trip, or some other casualty result.

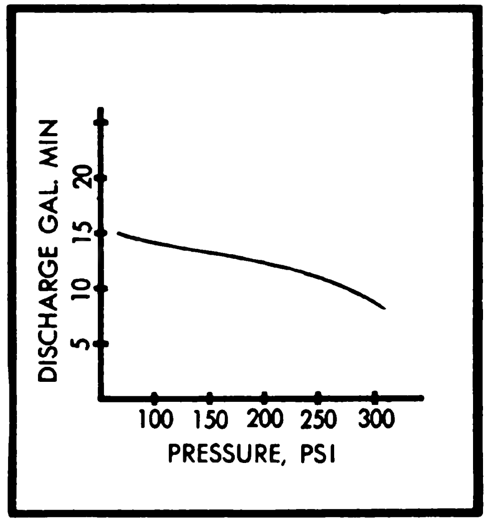


Figure 177

When a rotary pump is operating at a constant speed, the amount of liquid delivered to the discharge line—the effective discharge—decreases slightly as the pumping head increases. The drop in discharge is due to leakage back into the suction line. The leakage increases with the head pumped against by an amount determined by the design of the pump, Figure 177.

Varieties. No attempt will be made to enumerate all the many varieties of rotary pumps. The rotor may be a simple cylindrical element which is made to pass around the pump chamber, hugging the wall to produce a seal and push liquid ahead of it; it may consist of elements terminating in two, three or more lobes; gears may be used as in Figure 176, but with almost any number of teeth, and of a variety of forms; elements looking like gears may be employed, although one of the pair may not be capable of rotating the other; screws may be used which carry the liquid through the pump in their hollowed-out channels; sliding or swinging vanes may be employed which form a seal with the walls of the chamber because they are pushed out from the center of the rotor by mechanical arrangements or are thrown out by centrifugal force; or small pistons moving in and out from the center of the rotor may be used to push liquid through the pump. Liquid may enter and leave the pump at the outside border of the chamber, or it may enter and leave from a point near the center of the rotor.

Fields of application. The rotary pump has an extremely wide range of application, and has superseded the reciprocating pump

in numerous industrial operations. The advantages of the rotary pump over the reciprocating pump are simpler construction, smaller dimensions for a given capacity, direct drive from the electric motor removing reciprocating difficulties, and the delivery of a more continuous flow practically free from pulsations. The rotary pump also has the advantage of being self-priming and of being capable of operating against high suction lifts. Installation and maintenance costs are small.

Some designs of rotary pumps are especially adapted to handling highly viscous liquids, such as asphalt, heavy lubricating oils, tars, pitch, and heavy fuel oils. These pumps are also widely used to handle liquids of low viscosity such as gasoline, benzine, and light fuel oil. Liquids that might be affected by severe agitation are handled satisfactorily by rotary pumps, since they move liquid smoothly in comparison to centrifugal or reciprocating pumps.

Typical Rotary Pumps: Gear Pumps

The gear type is the most common variety of rotary pump. Three kinds of true gears are used: spur, helical and herringbone, which are illustrated in Figure 178. At the center is shown the ordinary spur gear, while at the top and bottom stand respectively helical and herringbone gears. A helix is the curve produced when a straight line moves up or down around the surface of a cylinder. A herringbone is composed of two meeting helixes going in opposite directions.

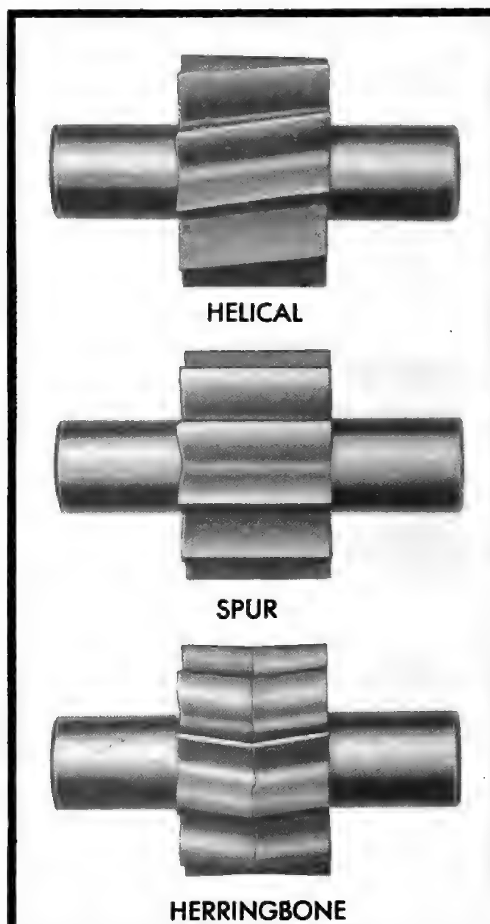


Figure 178

Spur and helical gear pumps are generally reversible in action, while herringbone gear pumps are not. When the latter are rotated in the wrong direction, oil is held in the middle "*v*" and the resultant back pressure strains the shaft and bearings.

Spur gear pumps. These are made by a large number of manu facturers, and are similar to Figure 176, although the gears may have almost any number of teeth. Pumps of this kind usually are run relatively slowly—at speeds under 500 revolutions per minute —and are of fairly small capacity. Thus a Worthington fuel oil service pump with spur gears used by the Navy has a capacity of 12 gallons per minute with a pump speed of 460 revolutions per minute, and can discharge this volume against a pressure head of 400 pounds per square inch.

Helical gear pumps. Figure 179 shows a helical gear pump with an associated valve block. The liquid is carried up around the wall

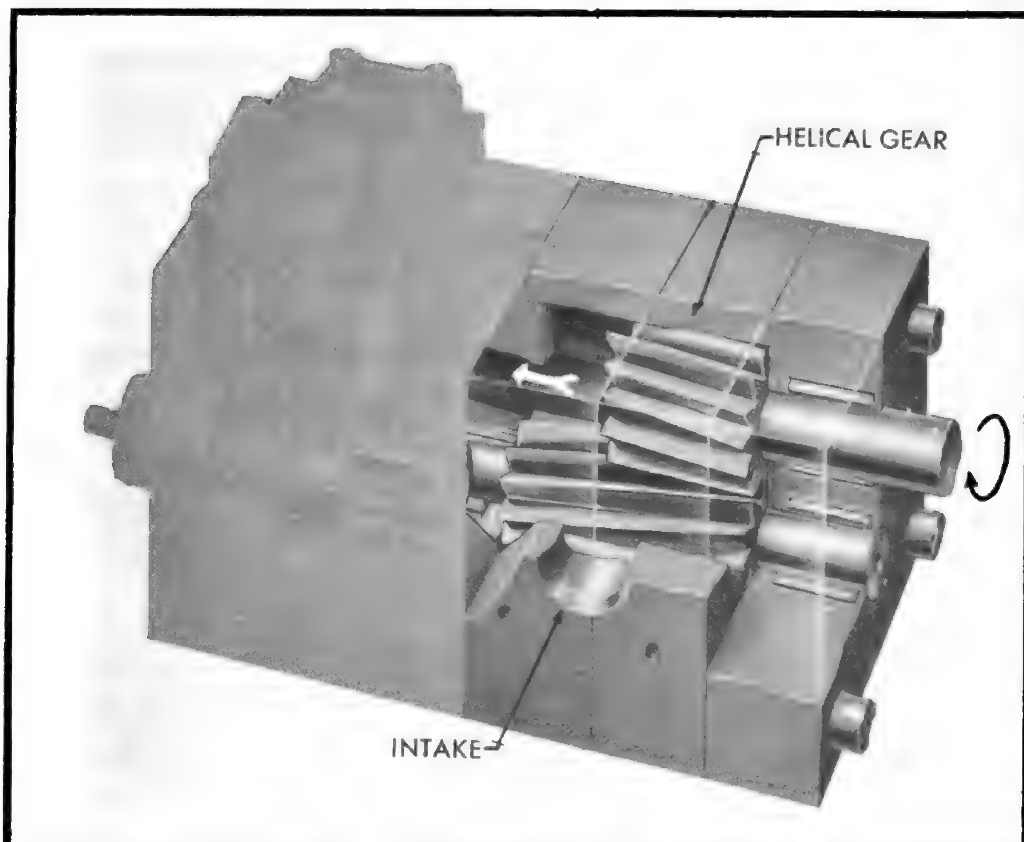


Figure 179

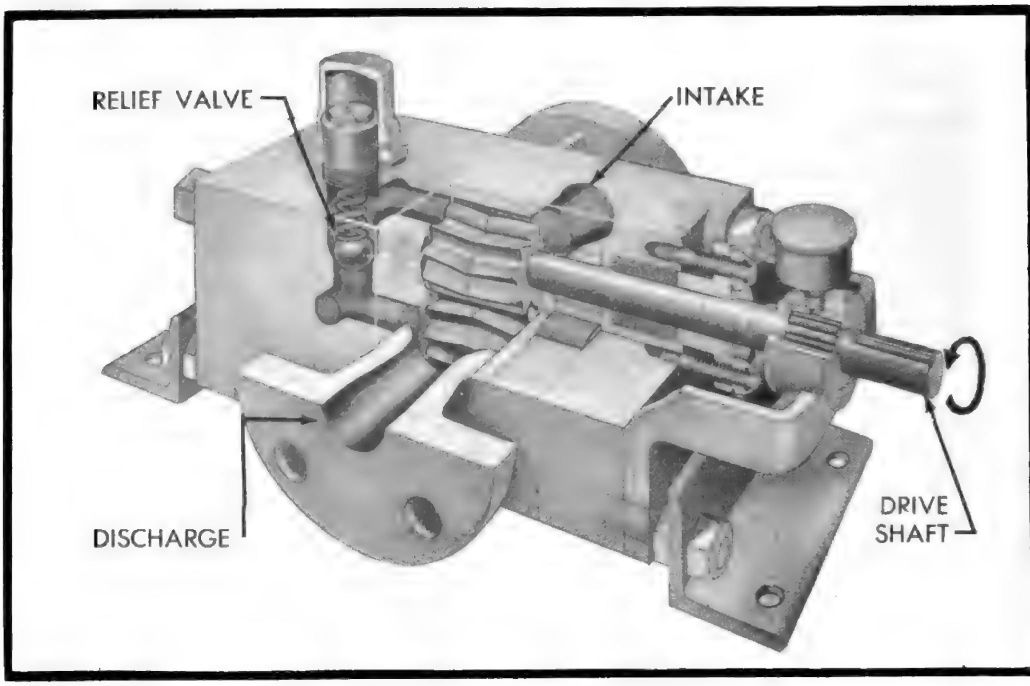


Figure 180

of the pump chamber, in exactly the same manner as with the spur gear pump, and is pushed out at the discharge side by the meshing of the gears. Since helical gear pumps run very quietly at high speeds, and can deliver relatively large volumes of liquid, they are well adapted for use in operating machines.

Herringbone gear pumps. Figure 180 shows a herringbone gear pump. The liquid is pumped in the same manner as before, but the relatively larger space at the center of the gears tends to minimize pulsations in the discharge. In addition end thrust is in considerable measure equalized by the opposed helical gears forming the herringbone.

Northern Series 4000 nitralloy gear pumps. The Northern Pump Company has developed a pump whose casing can be assembled from stock parts in the form of metal blocks to give a wide variety of arrangements to fit specific conditions. A set of such blocks is shown in Figure 181, while Figure 182 shows one-half of the body of a herringbone gear pump constructed out of these blocks. The pump shown in Figure 180 was also assembled out of these blocks. Five blocks are required to make a pump, as shown in Figure 181.

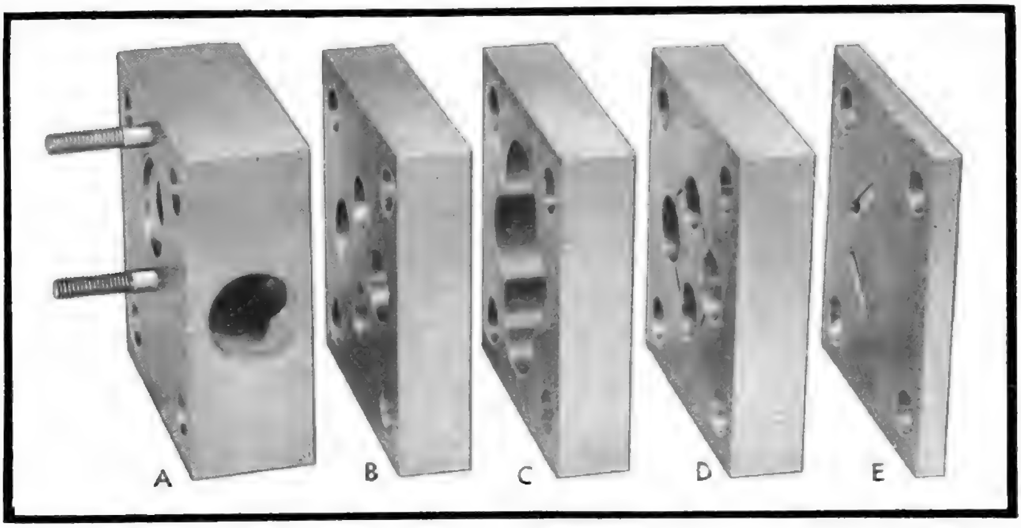


Figure 181

Block *A* is an end plate containing the intake and discharge parts (only one being shown in this illustration) and the shaft packing gland. Block *B* and *D* contain the shaft bearings. Block *C* contains the cylinders within which the gears revolve. Block *E* is an end plate. Four studs with nuts to hold the blocks together complete the casing assembly.

Into these blocks, or others of different internal construction, are fitted the proper gears and drive shaft, or the appropriate valves for some particular function. The pump may be provided with any

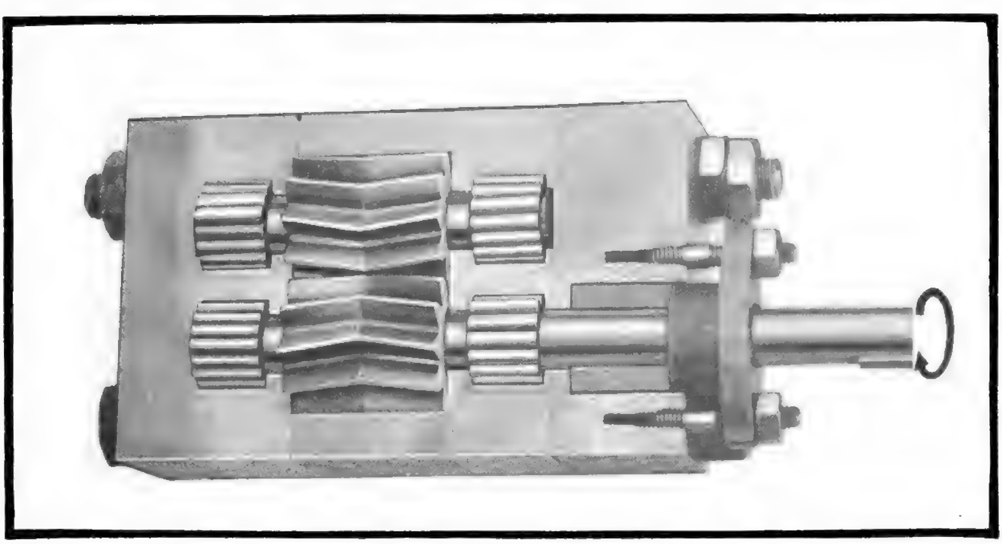
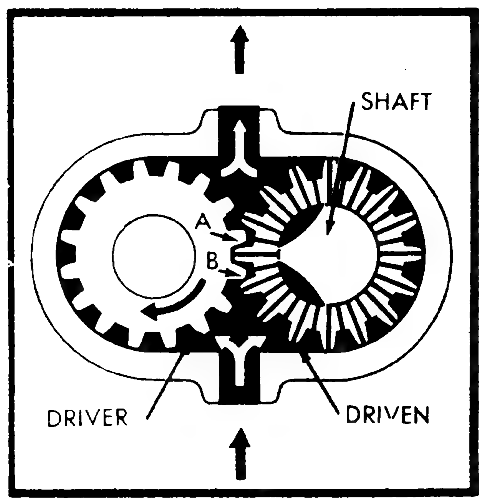


Figure 182

of the three kinds of gears. No gaskets are necessary between the blocks, since the surfaces are accurately machined.

The whole pump is made of nitralloy steel, which is harder and stronger than carbon steel and very resistant to corrosive action. A fuel-oil booster pump, capable of delivering 200 gallons per hour, weighs only about 14 pounds. Pumps of this kind deliver up to 1000 pounds per square inch, or more for specially designed pumps, and have capacities from 1 gallon per hour to 100 gallons per minute. Pump speed varies from 690 to 1750 revolutions per minute.



The Northern 4000 Series pump has been widely used by the Navy for diesel and fuel oil service, for lubricating oil service, as a steering gear pump, and as a windless pump, and for hoists, airplane elevators, catapults, training and elevating gears, rammers, etc.

This pump depends for lubrication upon the oil being pumped. the cylinders and gears and possibly even the shafting may be damaged in a few seconds if oil does not immediately enter the pump. Dirt or air in the system will tend to affect its efficiency. After long or severe service, leakage and vibration may develop due to wear, so that the old parts may have to be replaced.

Barnes gear pump. In the ordinary gear pump a certain amount of liquid will be trapped between the teeth of the returning gears at the point where they mesh, even when the pump is new and clearances are still very close. As the pump wears, this trapped liquid may create a major problem, since it sets up a strong pressure opposed to the action of the pump and tending to spread the gears. The Barnes gear pump, as shown in Figure 183, is constructed

with small passages running through and between the teeth of the driven gear. This gear rotates around a stationary shaft having two recesses which are arranged so that trapped liquid is forced through the passages into the recesses and out into either the discharge or the inlet area through other passages not at the moment in mesh at the center.

In Figure 183, liquid caught at point *A* will be driven through one recess in the stationary shaft out into the discharge, while liquid is also free to fill the recess under *B* and relieve the vacuum that would otherwise form between the gears as they unmesh. The position of the central shaft on these pumps also can be adjusted so that some portion of the liquid trapped between the meshing gears will be returned to the inlet area, thus giving variable delivery. Discharge can be reduced about one-third by this method.

In a variation, a quite different result can be secured. The trapped liquid can be delivered to a second outlet port running through the center of the shaft (Figure 184). By slightly shifting the shaft in one direction or the other, flow from the second outlet can be altered in quantity, and if desired can be returned to the inlet side of the pump, thereby in some degree giving variable delivery through the main outlet. Or flow through the shaft can be sent somewhere in the system to do work. Flow through the shaft can be varied from about 10 to 30 per cent of the pump's capacity.

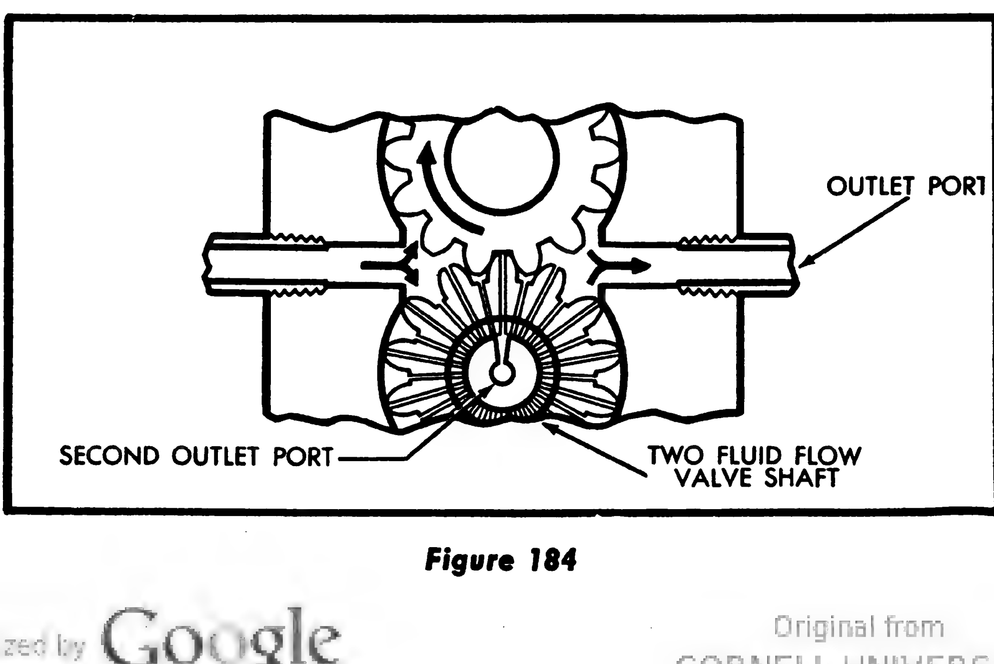


Figure 184

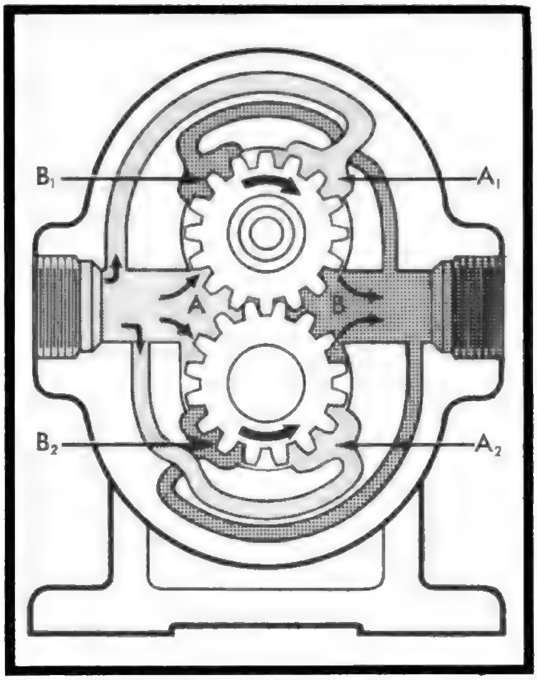


Figure 185

Vickers balanced gear pump. The ordinary gear pump is hydraulically unbalanced. Liquid enters on one side of the pump and leaves by the other, so that the pump parts are subjected to an unbalanced force opposed to the direction of flow, since pressure at the discharge is greater than at the inlet. Vickers, Inc., have developed a gear pump that is in balance, since it is constructed with pairs of opposed inlets and outlets, as shown in Figure 185. Liquid enters the pump chamber not only at A but also at A_1 and A_2, on the opposite side of the chamber, producing a balance of inlet pressures. Similarly, liquid from the discharge of the pump is free to reach B_1 and B_2, opposite to B, producing a balance of outlet pressures. Liquid entering the pump chamber at A leaves by way of B. Little or no liquid enters at A_1 or A_2, or leaves at B_1 or B_2. The pump can deliver pressures up to 1000 pounds per square inch, and the direction of delivery can be reversed by changing the direction of rotation of the drive shaft.

This pump is often used as a hydraulic motor as well as a pump proper. When so used, liquid is delivered to it by another pump, which is itself driven by a source of power, and this liquid drives the gears of the balanced gear pump, providing power for a work operation. By changing the volume of liquid delivered to the hydraulic motor, the speed of rotation of its output shaft can be altered. The hydraulic motor can be set up anywhere that a system of pipes can carry the driving liquid, giving valuable flexibility in the organization of the work system. The unit is also very compact. Power to run it can be steadily and economically maintained.

Gear-Like Pumps

Kinney heliquad pumps (Figure 186). These pumps make use of gear-like rotors that are not actually gears, since they are not capable of driving each other. One rotor must be powered by a timing gear. Each of the elements is composed of from three to nine meshing parts, which may be of helical or herringbone construction. Figure 187 shows a variety of these rotors. The liquid is carried around the pump chamber in the outside pockets of the rotating elements, and is pushed out at the discharge by the meeting of these elements near the center of the pump.

Figure 186 gives a cut-away view of one of these pumps. One of the rotors is mounted on the drive shaft, and the other is driven by a set of timing gears (not shown). Liquid enters the pump chamber as indicated, and is carried around the chamber in the rotor spaces. It is prevented from passing directly through the center of the pump by the meshing of the rotors. The liquid is pushed out of the

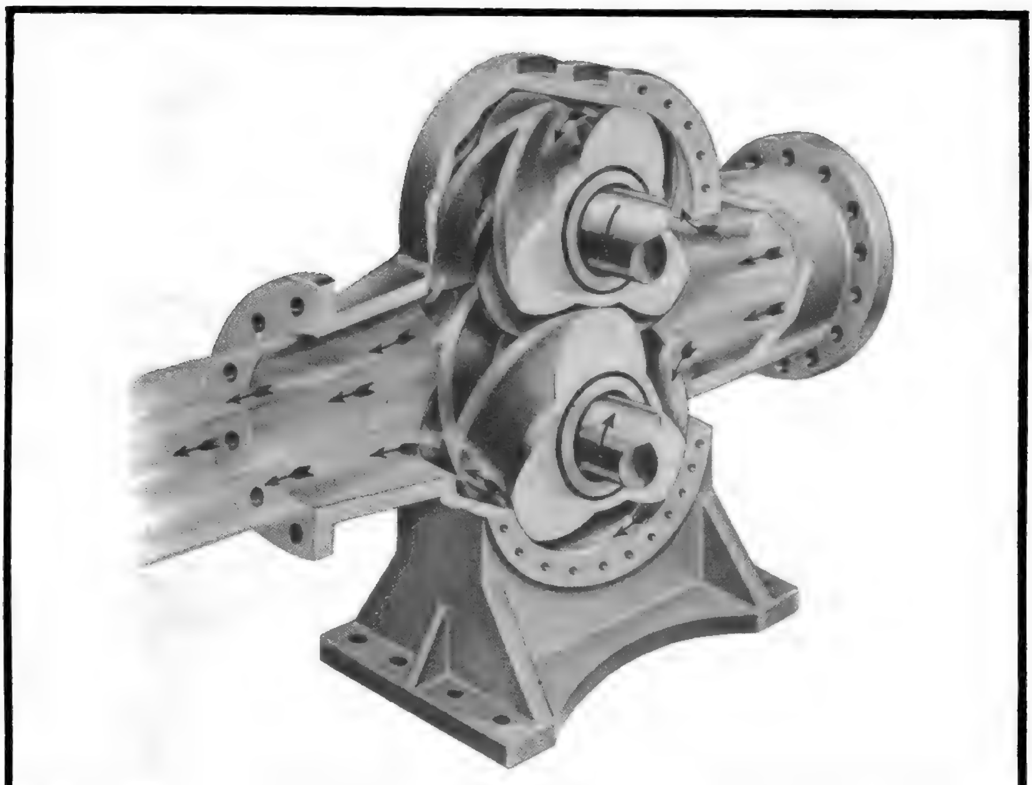


Figure 186

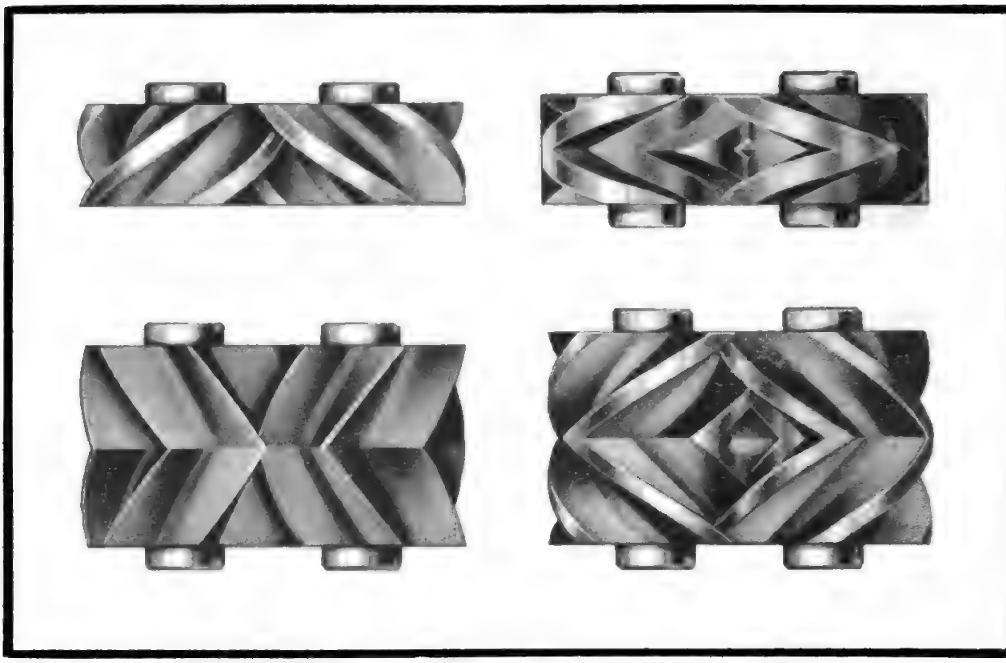


Figure 187

pump chamber as the rotating elements meet on the discharge side.

This pump is used for main lubricating oil service, as a fuel oil tank drain pump, and for a variety of miscellaneous services, including diesel oil supply and diesel engine cooling water service.

Screw Pumps

Quimby screw pump. This pump is shown in top and side views in Figure 188. Liquid from suction is introduced at both ends of two meshing screws, and is carried to the discharge, at the center, by their rotation. The screws are rotated, one directly by the source of power and the other through a set of timing gears.

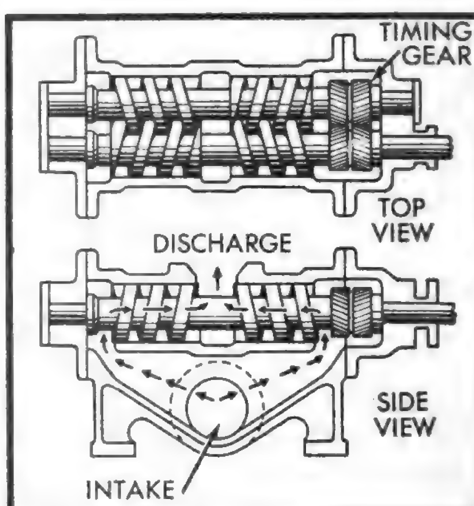


Figure 188

This pump is used for boiler fuel oil service, and for main lubrica-

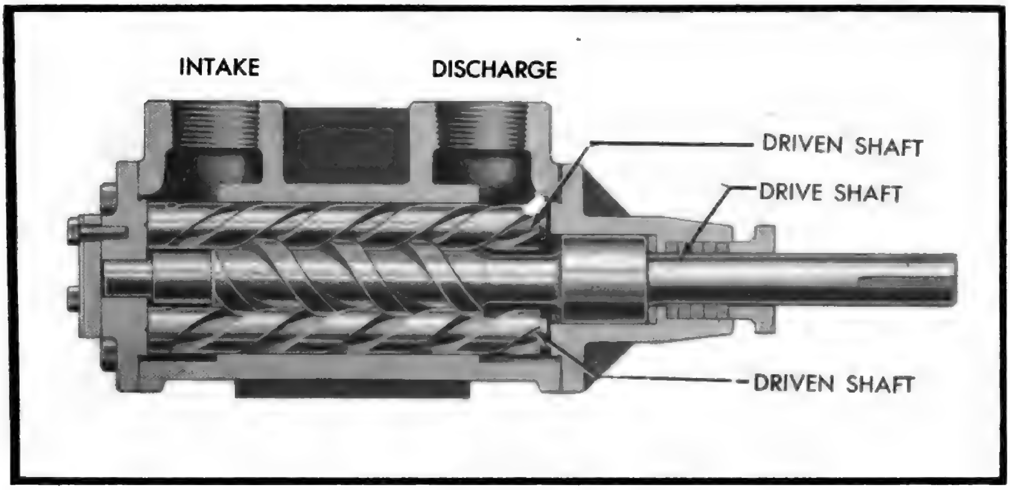


Figure 189

ting oil service for surface vessels. It is well adapted to handle oils with a considerable amount of entrained air, which in some pumps would lead to considerable vibration.

De Laval IMO pump. Here we have a power rotor meshing with two opposed screws. The liquid may enter at the far end of the pump from the prime mover, and leave at the near end, as shown in Figure 189, or the screws may be so constructed that liquid enters and leaves from opposite sides of the center, as shown in Figure 190, thereby balancing end thrust due to the pressure of the liquid.

This pump is used for fuel oil service, as a fuel oil booster and fuel oil transfer pump, as a lubricating oil pump, and for miscellaneous services. With a cast iron case the pump will take pressures up to 200 pounds per square inch, or up to 500 pounds per square inch with a semi-steel case. The pump requires but one packing box and is otherwise simple in construction.

Lobe Pumps

Figure 191 shows a two-lobe pump. The principle of action is exactly the same as for the gear pump already described. The two elements are rotated, one being directly driven by the source of power, the other through timing gears. As the elements rotate, liquid is caught between each lobe and the wall of the pump chamber, and

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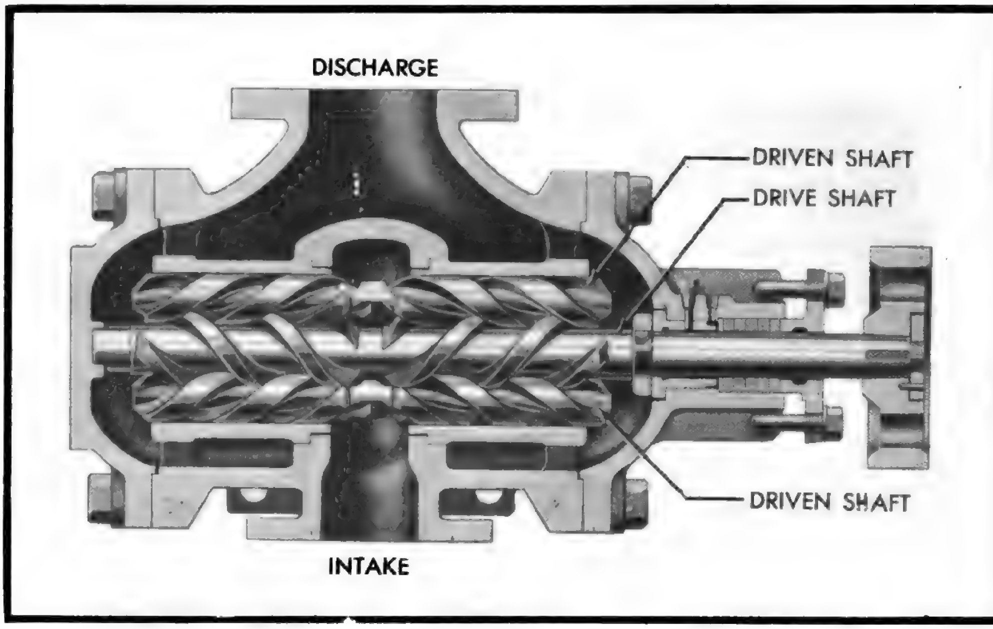


Figure 190

carried around from suction to discharge. As each volume of liquid leaves the intake chamber, the pressure there is lowered, and additional liquid is forced up into the chamber from the source of supply. The lobes are so constructed that there is a continuous seal at their point of juncture at the center of the pump chamber.

Northern cycloid pump, Series 3000 (Figure 192). This pump is fitted with two three-lobed rotating members, sometimes incorrectly

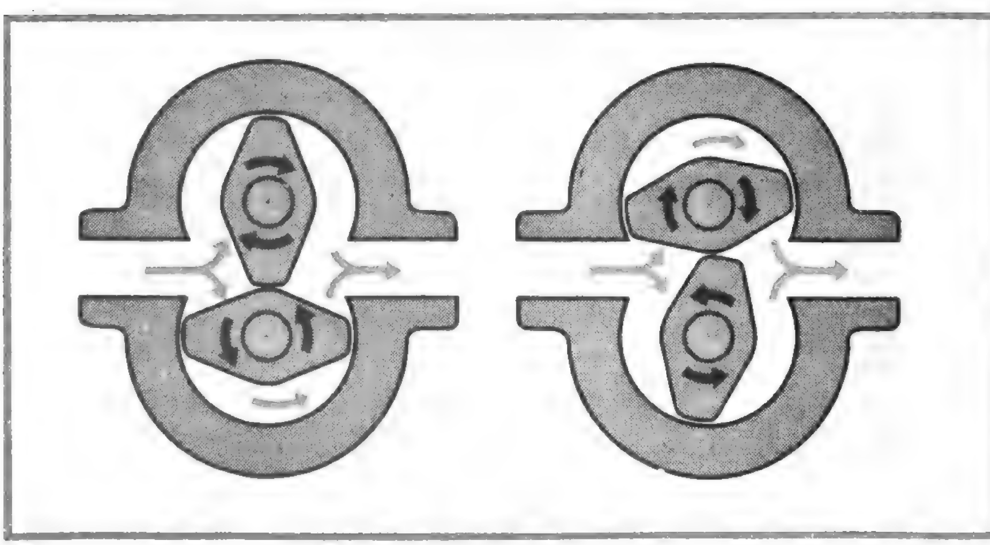


Figure 191

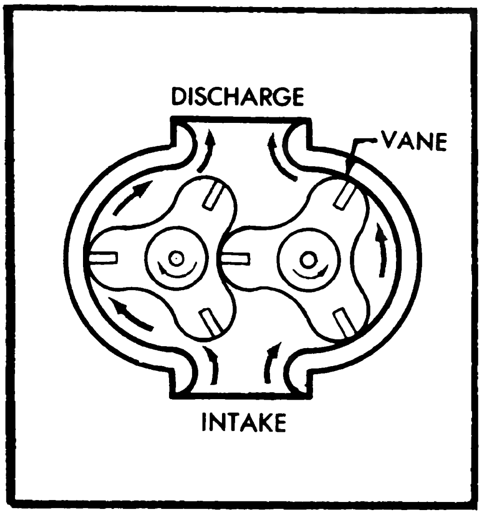


Figure 192

called impellers. One of the rotating members is directly driven by the source of power, while the other member is driven by a timing gear which stands outside the pump chamber. As will be noted in Figure 192, the lobes are fitted with small vanes at their outer edges to improve the seal of the pump. Although these vanes are mechanically held in their slots, they are to some extent free to move outwards. Centrifugal force keeps them snug against the wall of the chamber and the other rotating member. Cycloid pumps are made in a number of sizes, designed to rotate at from 300 to 600 revolutions per minute and to deliver from 46 to 627 gallons per minute at pressures of from 50 to 250 pounds per square inch, depending on the pump. The Northern cycloid pump is used to drain sediment from fuel oil tanks, and for similar uses. It is not adapted to high-pressure fuel oil service because its large rotors give a heavy bearing load on the driving motor.

Rotating Plunger Pumps

Kinney rotating plunger pump (Figure 193). In this pump a plunger is set off-center on a drive shaft which is rotated by the source of power. The plunger is driven up and down and around the chamber by the rotation of the shaft in such a way as to make a moving seal with the walls of the chamber, from the inlet to the discharge. In doing so it alternately opens and closes a passage to the discharge, since an arm from the piston moves up and down in a slide pin to give this result.

Figure 193 shows this pump in three positions—with the plunger starting on the down stroke (*A*), with the plunger at its lowest point (*B*), and with the plunger near the top of its stroke (*C*). The direction of rotation in the illustration is counterclockwise. The

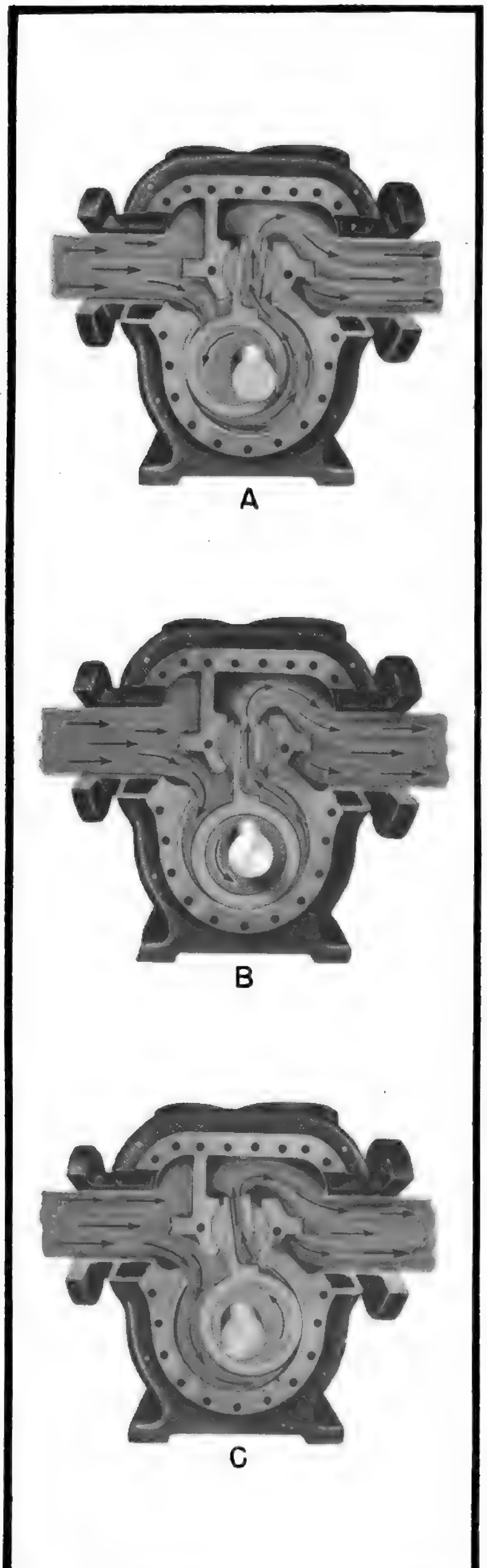


Figure 193

drive shaft is set in the middle of the pump chamber, with the plunger placed off-center on the shaft. The manner in which the discharge opening works should be clear from the drawings. The actual pump has two units of the kind illustrated, each working in its own compartment in the pump chamber but with coverging suction and discharge openings. The two pistons are set so that one is at the height of its discharge when the exit of the other piston is completely closed, so that the pump gives very nearly uniform flow.

Rotating plunger pumps can carry extremely viscous liquids such as asphalt and molasses, or mixtures of viscous liquids and gases. They can produce an extremely large suction lift—30 feet or more against atmospheric pressure. Model SD works under rather low pressures, usually no more than 150 pounds per square inch, but Model SP which the manufacturer developed originally for Navy high-pressure fuel oil service, can provide a pressure of 400 pounds per square inch.

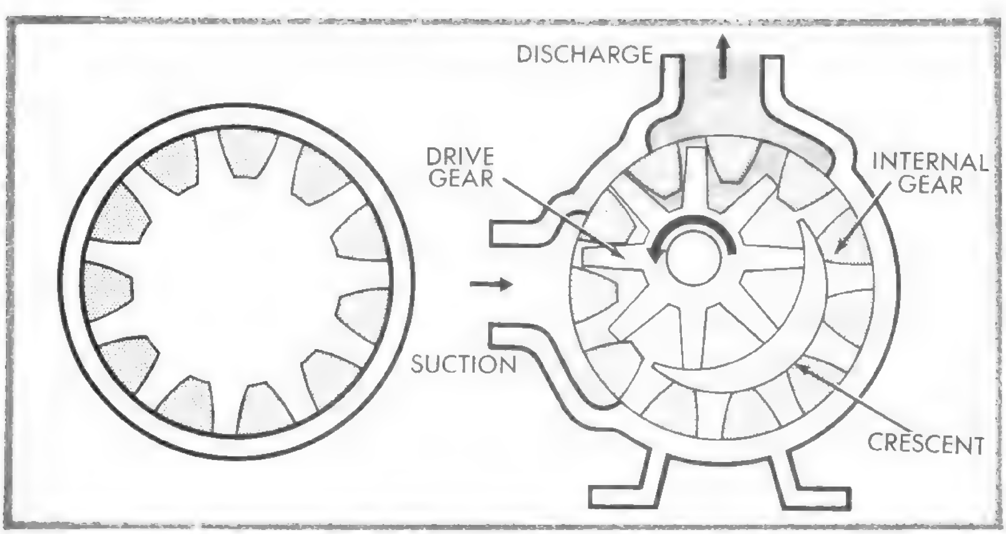
The pump is used for boiler fuel oil service, as a fuel oil booster and fuel oil transfer pump, and also to drain sediment from fuel

oil tanks. Pumps performing these last three services are required to pump oil which may contain sand, sediment and salt water through long lengths of suction piping which may have a number of pockets or traps, and the oil pumped may be of very high viscosity or at a high temperature, so that it vaporizes easily. Only a valveless pump could give satisfactory service under such difficult conditions.

Since these pumps are lubricated by the liquid pumped, they should not be operated empty. They should be started with a reduced load and gradually increased to the specified working level. The pump can easily be disassembled, but care must be taken in reassembling to see that the hollow part of the slide stands on the discharge side of the pump. The piston should be replaced with the mark "suction" facing the intake side.

Internal Gear Pumps

In ordinary gear systems, both sets of teeth project outwards from the centers of the gears. In an internal gear system, the teeth of one gear project outwards, but the teeth of the other gear project inwards toward the center of the gear, as shown in Figure 194. One gear wheel stands inside the other in an internal gear pump.



Figures 194 and 195

A gear of this kind can rotate, or be rotated by, a suitably constructed companion gear.

Internal gear pumps require an oil filter in the inlet line, or the close meshing parts will soon wear or stick.

Northern internal gear pump. The manner in which this pump is constructed is shown in Figure 195. A gear directly attached to the drive shaft of the pump is set off-center in a circular chamber fitted around its circumference with the spurs of an internal gear. The two gears mesh on one side of the pump chamber, between the suction and the discharge. On the opposite side of the chamber a crescent-shaped form stands in the space between the two gears in such a way as to provide a close clearance with them.

The rotation of the central gear by the shaft causes the outside gear to rotate, since the two are in mesh. Everything in the chamber rotates except the crescent, causing liquid to be trapped in the gear spaces as they pass the crescent. This liquid is carried from the suction to the discharge, where it is forced out of the pump by the meshing of the gears. As liquid is carried away from the intake side of the pump, the pressure there is diminished, and other liquid is therefore forced in. The action, in other words, is exactly like that of all other gear pumps. The direction of flow in this pump can be reversed by shifting the position of the crescent 180°.

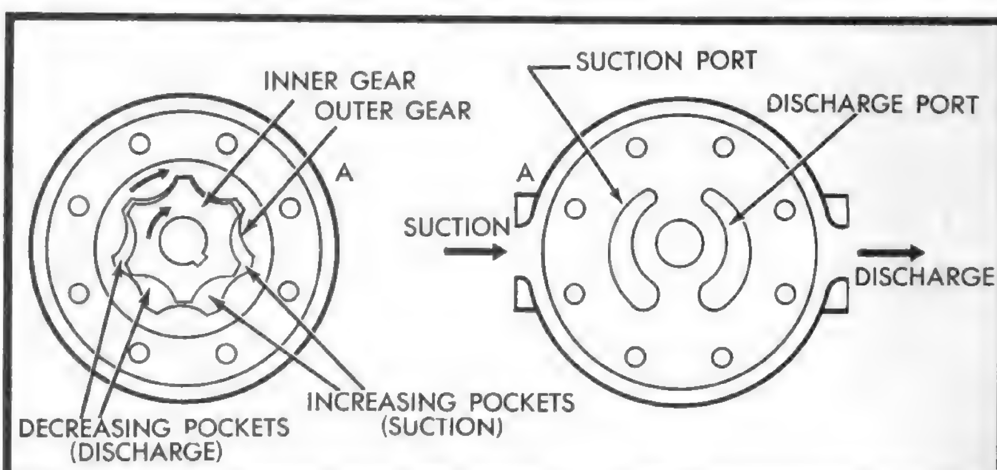


Figure 196

Gerotor pump. An internal gear pump of quite another construction is manufactured by the May Oil Burner Company. The Gerotor mechanism consists of a pair of gear-shaped elements, one within the other, mounted in a pump chamber (Figure 196). This inner gear is directly connected to the drive shaft of the source of power, and itself drives the outer gear.

How the gears operate to give a pumping action is shown in Figure 197. The inner gear has one fewer teeth than the outer gear. The tooth form of each gear is related to that of the other in such a way that each tooth of the inner gear is always in sliding contact with the surface of the outer gear. Each meshing pair of teeth of the two gears definitely engages at just one place in the pump. In the drawings this place is at the top, at *X*, so that tooth *1* will be in mesh with tooth *2*, tooth *3* in mesh with tooth *4*, and so on, when each pair reaches position *X*.

At one side of the point of mesh, pockets of increasing size are formed as the gears rotate, while on the other side the pockets decrease in size. The pockets of increasing size are suction pockets, while those of decreasing size are discharge pockets. In Figure 197 it will be noted that the pockets on the right-hand side of the drawings are increasing in size as one moves down the page, while those on the left-hand side are decreasing in size. The intake side of the pump would therefore be to the right, and the discharge to the left. In Figure 196, since the right-hand side of the drawing was turned over to show the ports, intake and discharge appear reversed. Actually *A* in the one drawing covers *A* in the other.

The two gears travel slowly with respect to each other, even when the drive shaft is rotating rapidly. Thus in one pump the Gerotors revolve at 200 revolutions per minute with respect to each other when the pump shaft is rotating at 1800 revolutions per minute. These pumps can deliver from 0.4 to 30 gallons per minute at a pressure of 1000 pounds per square inch.

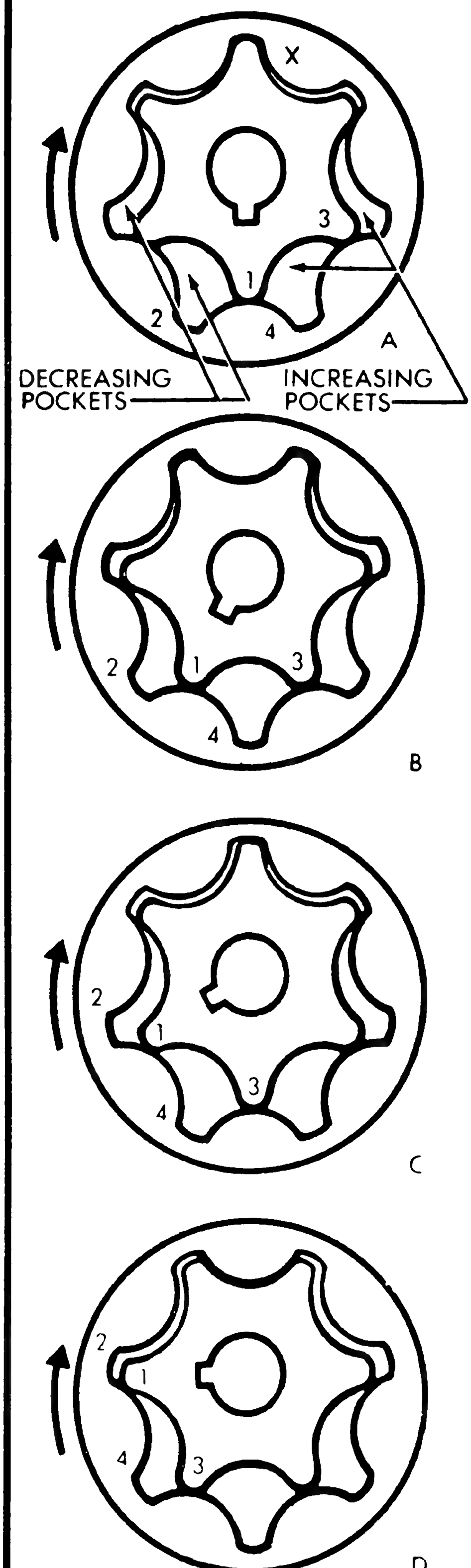


Figure 197

The Navy uses the Gerotor pump to supply oil to boilers for starting fires, as an element in fog-making devices, and in hand operated units to open the bow doors of landing craft, as likewise for the emergency operation of steering gear.

Vane Pumps

Vickers vane pump (Figure 198). In pumps of this kind, the rotor is fitted with a number of slots into which movable vanes are fitted. As the rotor revolves the vanes are thrown out by centrifugal force to bear against the surface of an oval shaped ring. They follow the contour of this surface which is machined into the inside wall of the ring. In addition to centrifugal force, liquid pressure from liquid which has leaked behind the vanes also holds them against the surface.

Figure 198 shows the general structure of this pump. It will be noted that the pump is balanced by virtue of the opposed pairs of inlets and outlets. The vanes scoop up liquid at each inlet and deliver it at the next outlet.

This pump delivers from 1 to 44 gallons per minute at a maximum discharge pressure which is usually 1000 pounds per square

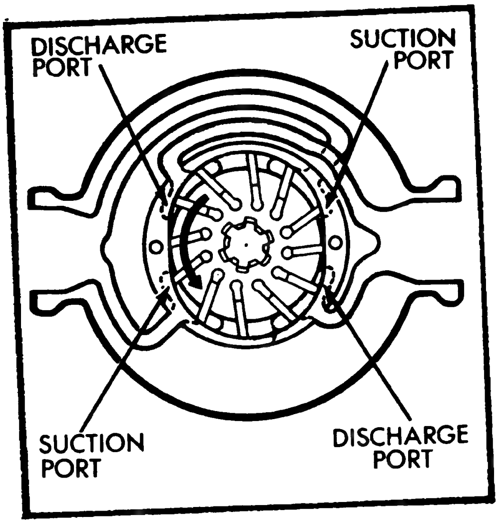


Figure 198

inch, and with a speed of from 850 to 1750 revolutions per minute.

A typical Navy use for the Vickers vane pump is for projectile and powder hoists for 5″/38 Cal. guns.

Racine vane pump (Figure 199). This pump is similar to the Vickers vane pump in some respects, but has a feature which permits adjustment to produce variable delivery. The rotor is set in a pressure-chamber ring which can be adjusted to stand off-center to the rotating element. As the degree of eccentricity is changed, a variable volume of liquid is pumped. As shown in the diagram, the vanes pump liquid from both ends, receiving the liquid on the left-hand side of the pump chamber and pushing or sweeping it out on the right-hand side for counterclockwise rotation.

The position of the pressure-chamber ring can be changed by hand by screwing down the adjustment mechanism shown in Figure 199, electrically, or hydraulically by means of an adjustable spring-set governor as shown in Figure 200. In the last case, the spring of the governor is set to maintain the desired pressure, and is held in balance by liquid admitted to its under surface. If the liquid pressure of the system increases above the set pressure of the spring, the spring will rise, permitting the pressure-chamber ring to take a position less off-center with respect to the rotor.

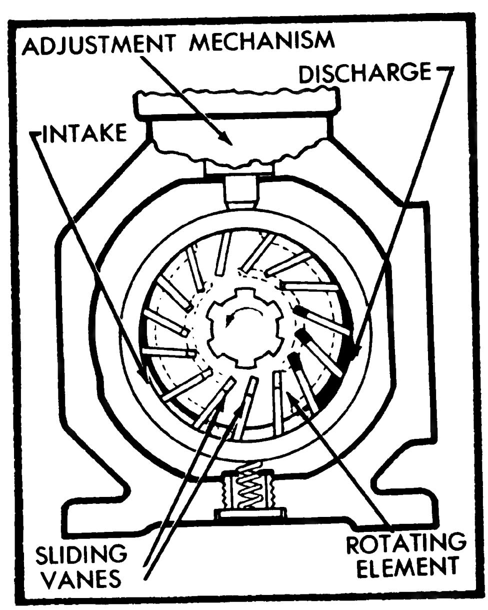


Figure 199

The pump delivers less liquid, and the pressure is lowered to the desired level. Adjustment of pressure at any level from 50 to 1000 pounds per square inch is possible.

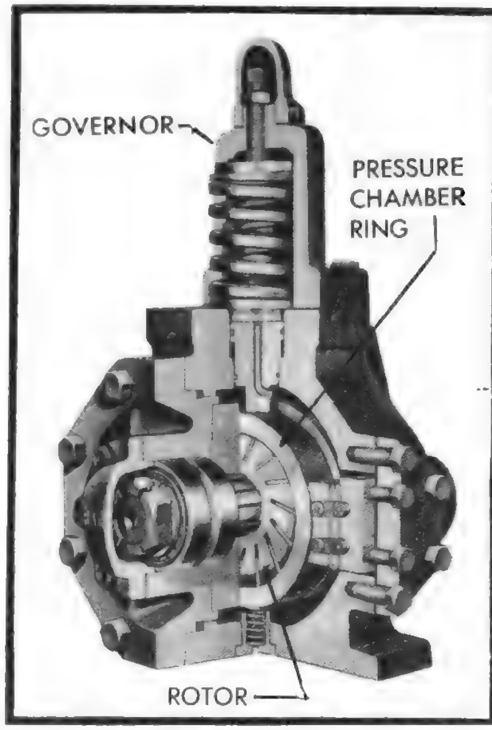


Figure 200

Installation and Operation

Rotary pumps, like all others, must be properly installed. The pump and the source of power must be correctly aligned, and all clearances must be as indicated by the manufacturer. The pump must be installed so that rotation will carry liquid from the inlet and deliver it to the outlet, since the action of many rotary pumps is not reversible.

All idle pumps should be turned daily by hand, and should be run under power at least once a week. All valves and other governing parts should be worked by hand at least once a week. The pump should be thoroughly cleaned at least once a year, and all parts checked for clearance and wear. Whenever repairs are made on a pump, a record should be kept so that it will be possible to judge the success with which the pump is performing its functions.

If the pump fails to deliver liquid, or delivers too little, there may be an obstruction in the suction line (including an air pocket), the pump's parts may be badly worn, or the packing defective. It is assumed that the suction lift is not too great, that the pump is set to rotate in the right direction, and that it is pumping a suitable type of liquid at the proper speed. If the pump requires too much power, the speed may be too high, the liquid too viscous for the pump, the suction or discharge lines obstructed, the driving shaft bent, the rotating element or the stuffing boxes too tight, or the pump misaligned. If the action of the pump produces water

hammer, the liquid being pumped may contain air or gas, the suction line may be leaky, the velocity of entrance into the pump may be too great, or the suction line may not be sufficiently direct. The remedy for each of the above conditions is obvious.

The general instructions on care and maintenance of pumps given in Chapter 8, as well as the more detailed instructions given in Chapter 47 of the *Bureau of Ships Manual* (Section III, parts 2-5), should be familiar to those charged with the maintenance of positive displacement rotary pumps.

QUESTIONS

1. Identify: rotor; helix; herringbone; Gerotor; internal gear; timing gear.

2. What path does the liquid take in moving through a gear pump?

3. How is the Northern 4000 nitralloy pump constructed?

4. How does the Barnes gear pump handle the problem of liquid trapped in the returning gears?

5. How is hydraulic force balanced in the Vickers balanced gear pump?

6. Why is it possible to use the Vickers balanced gear pump as a hydraulic motor? How does a hydraulic motor differ from a pump?

7. What is the advantage of helical and herringbone gears over the ordinary spur type?

8. Why must each of the rotors in a Kinney heliquad pump be activated by the source of power?

9. Explain the action of the Kinney rotating plunger pump. For what types of use are these pumps particularly adapted?

10. Explain the construction of the De Laval IMO pump.

11. How is the pumping action of the Gerotor pump obtained?

12. What is the purpose of the crescent in an internal gear pump?

13. How do vane pumps operate?

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Chapter 11

RADIAL PISTON PUMPS AND MOTORS

This chapter turns to pumps in which the amount of liquid delivered can be varied while the pump is running at a constant speed —without change, that is, in the speed of the source of power. The amount of delivery can be varied from zero to the full capacity of the pump, and the direction of movement at the point of work can be reversed. This chapter will deal with radial piston pumps and motors, while Chapter 12 will take up axial piston units.

In most of the systems so far studied in previous chapters we can control the action of the system only by means of valves. With variable delivery pumps, however, the speed and direction of the work operation can be controlled right at the pump, by changing the volume and direction of flow of the pump output. In radial and axial piston pumps volume is controlled by varying the length of the piston strokes of the pump. The direction of flow is controlled by reversing input and output relations in the pump. How these actions are accomplished will be explained in this and the next chapter.

Radial Piston Pumps

How they work. We know that a piston moving back and forth in a cylinder can draw liquid in and then push it out. We also know

that we can control the direction of flow of the liquid by the use of valves which permit us to draw liquid in from one pipe and push it out into another pipe, as shown in Figure 201. The same principle can be applied in a rotary pump, as study of Figures 202 and 203 will show. The pump consists of a *pintle* which remains stationary and is actually a valve, as we shall soon see; a *cylinder block* which revolves around the pintle and contains the cylinders in which the pistons operate; a *rotor* which houses the reaction rings of hardened steel against which the piston heads press; and a *slide block* which is used to control the length of the piston strokes. The slide block does not revolve but houses and supports the rotor, which does revolve due to the friction set up by the sliding action between the piston heads and the reaction ring. A drive shaft is secured to the cylinder block.

In Figure 202, suppose we have a liquid in space *X* in one of the cylinders of the cylinder block when the piston is at position *A*. If we rotate the cylinder block and piston in a clockwise direction, the piston will be forced into its cylinder as it approaches position *B*. The piston will thus drive a quantity of liquid out of the cylinder and into the outlet port above the pintle. This pumping action is due to the fact that the rotor, in the slide block, is off-center in relation to the center of the cylinder block.

In Figure 203 we see what happens after the piston has reached position *B* and has pushed its liquid out. The cylinder outlet will be blocked as the piston moves from position *B* to *C*. No pumping or suction will occur while the piston and cylinder are passing

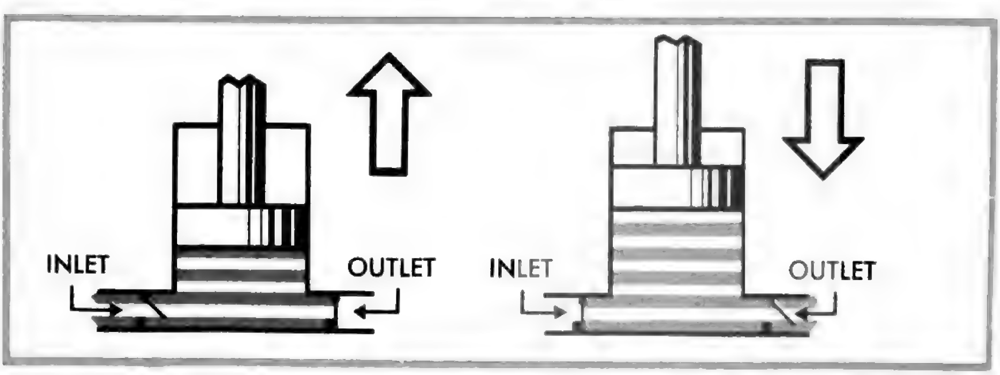

Figure 201

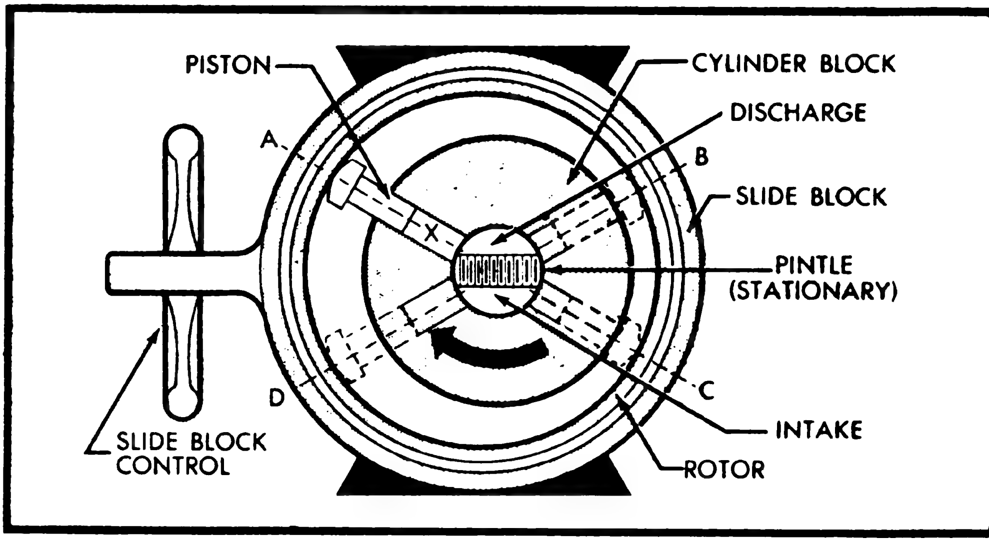

Figure 202

over the solid part or land of the pintle. After the piston has moved to position *C*, however, it can start drawing liquid into the cylinder as the rotation of the cylinder block moves the piston to position *D*, at which point the cylinder has taken on a full charge of liquid. The pistons are driven out of the cylinder block by centrifugal force as the cylinder block is revolved. As the piston moves across the pintle, no intake or discharge of liquid will occur. After the piston has passed the pintle *A* and starts again toward *B*, discharge will once more take place. Alternate intake and dis-

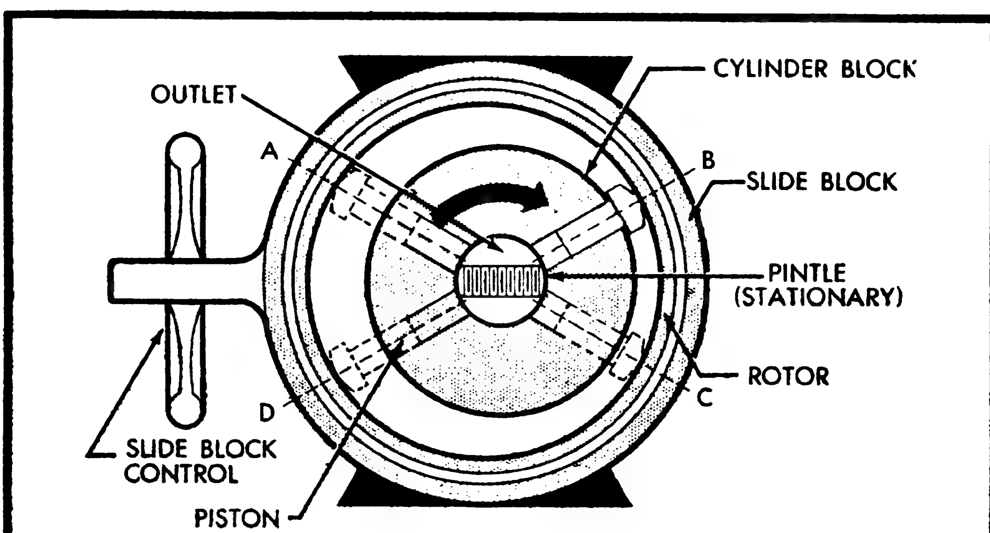

Figure 203

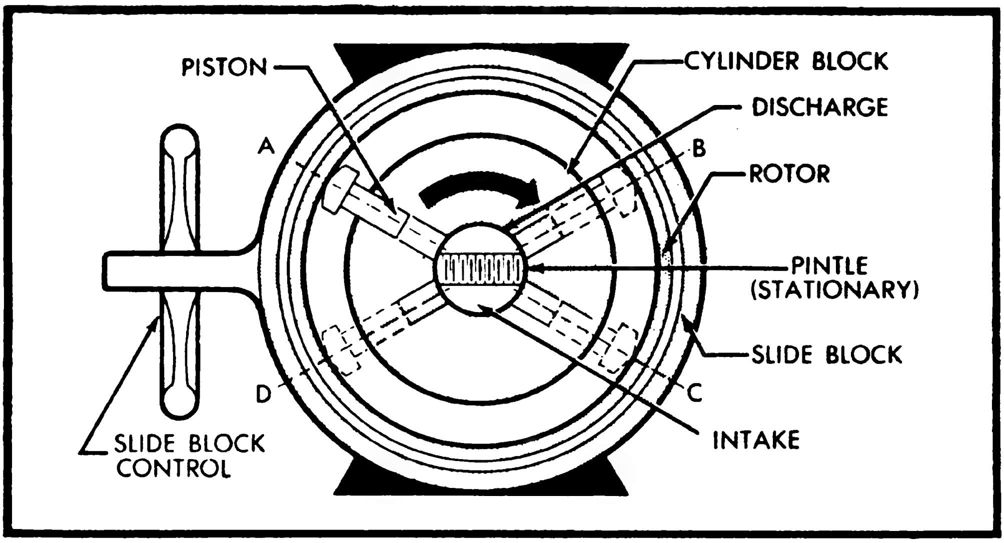

Figure 204

charge will continue as the rotor is revolved about its axis—intake on one side of the pintle and discharge on the other side, as the piston slides in and out. Its head will always be held against the reaction ring of the rotor by centrifugal force.

So far we have assumed that the center point of the rotor is different from the center point of the cylinder block. It is this difference of centers that produces the pumping action. If we move the rotor so that its center point is the same as that of the cylinder

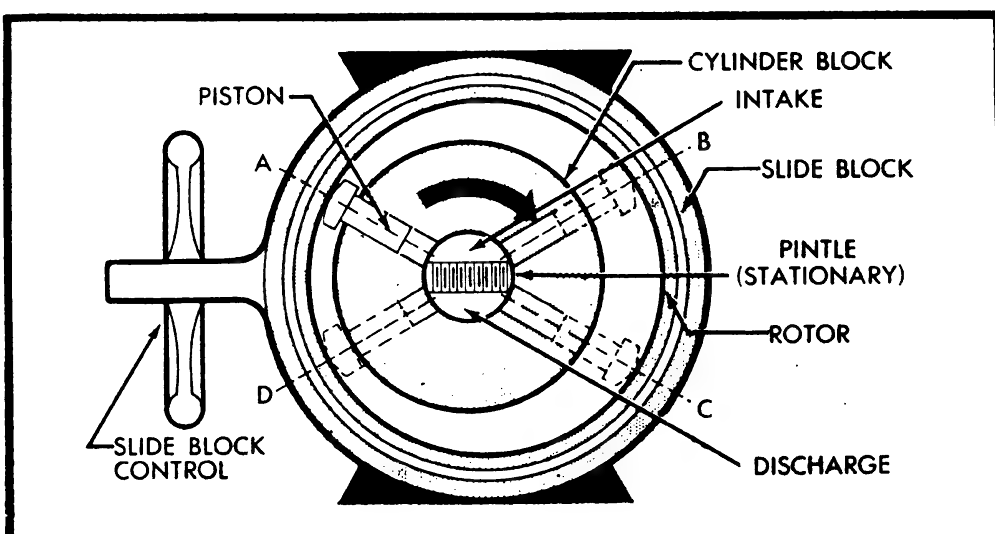

Figure 205

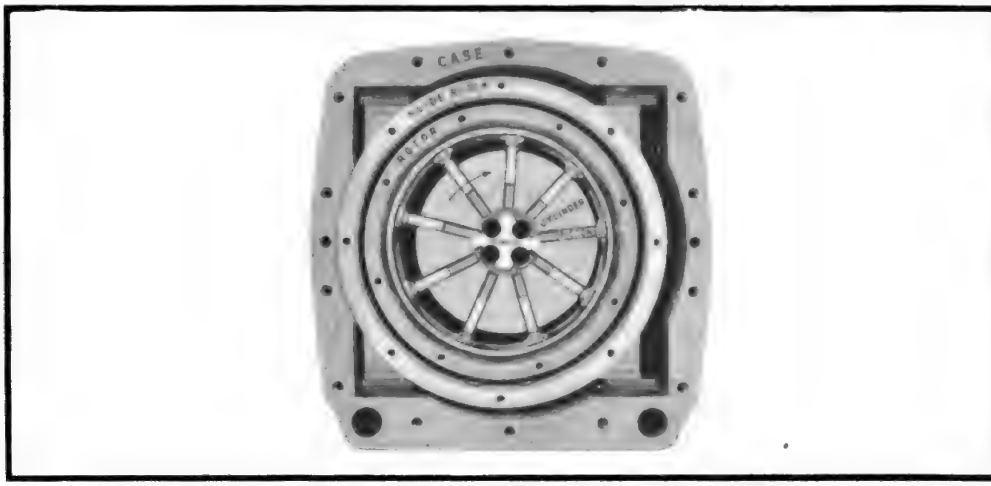

Figure 206

block, as shown in Figure 204, there will be no pumping action, since the piston does not move back and forth in the cylinder as it rotates with the cylinder block.

We can go one step further and reverse the flow in this pump, by moving the slide block, and therefore the rotor, to the right so that the relation of the centers of the rotor and cylinder block is reversed from the position shown in Figures 202 and 203. Figure 205 shows this arrangement. Now the piston will draw liquid in as it travels from *A* to *B* and will pump liquid out as it travels from *C* to *D*, reversing the flow of the liquid as shown in Figure 203.

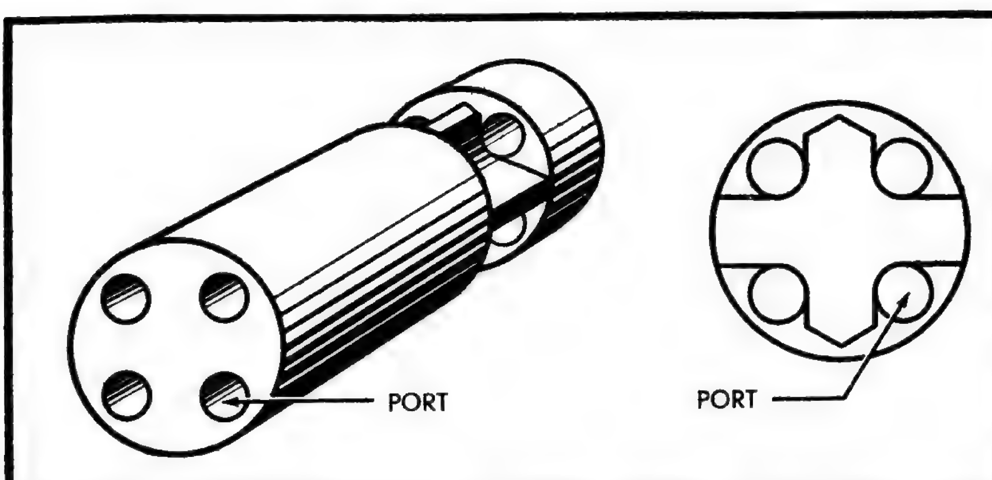

Figure 207

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So far in our illustrations the rotor has been shown in an extreme left or right position in relation to the cylinder block. The amount of adjustment or the difference in distance between the two centers determines the amount of liquid that will be pumped. This difference determines the length of the piston stroke, which controls the amount of liquid flow in or out of the cylinders.

A pump with only one piston would not be practical. But we can put a number of pistons in the same cylinder block (Figure 206). It will be noted that there is an odd number of pistons. This is because no more than one piston will be completely blocked at any one time by the pintle. If there were an even number of pistons spaced evenly around the cylinder block, for example eight, there would be occasions when two of the pistons would be blocked by the pintle, while at other times none would be blocked. This would mean that three pistons would be discharging at some time and four at others, causing a pulsation in flow. With an odd number of pistons spaced evenly around the cylinder block, only one piston is completely blocked by the pintle at any one time. This markedly reduces pulsation.

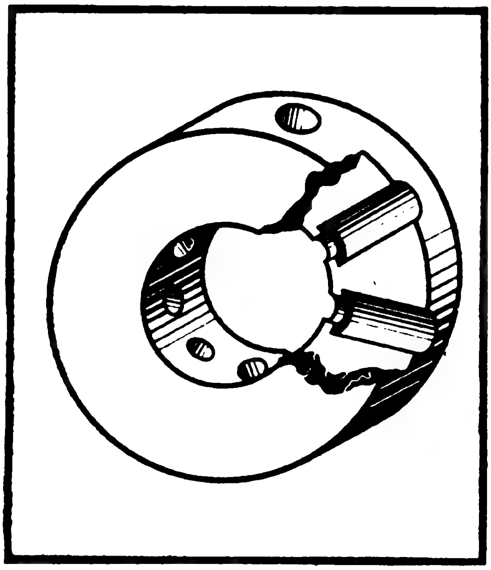

Figure 208

Parts of the pump: the pintle. In our first illustrations the pintle was shown for the sake of simplicity as a flat bar around which the rotor turns, and we made no attempt to collect the pumped liquid so that it could be put to useful work. Actually the pintle is a round bar which serves as a stationary shaft around which the cylinder block turns. The pintle shaft as shown in Figure 207 has four holes bored from one end lengthwise through part of its length, a pair each for intake and discharge. Two slots are cut in the sides of the shaft so that each slot connects two of the length-

wise holes. The two slots are in line with the pistons when the cylinder block is assembled on the pintle. One of these slots provides the path by which the liquid passes from the pistons to the two discharge holes bored in the pintle. The other slot connects the two inlet holes to the pistons when they are drawing liquid in. The discharge holes are connected through appropriate fittings to a discharge pipe so that liquid under pressure can be directed away from the pump. The other pair of holes is connected to an inlet pipe.

The cylinder block. This is a block of metal with a hole bored through its center to fit the pintle and with cylinder holes bored, equal distances apart, around its outside edge. The cylinder holes connect with the hole that receives the pintle. The relationship for one kind of pump is shown in Figure 208. Though cylinder blocks may differ in detail, they all serve the same purpose. Some appear to be almost solid, while others have spokelike cylinders radiating out from a center, as shown in Figure 209, which illustrates the Hele-Shaw cylinder block. Both cylinder holes and pintle holes are very accurately machined so that liquid loss around the piston or pintle will be kept to a minimum.

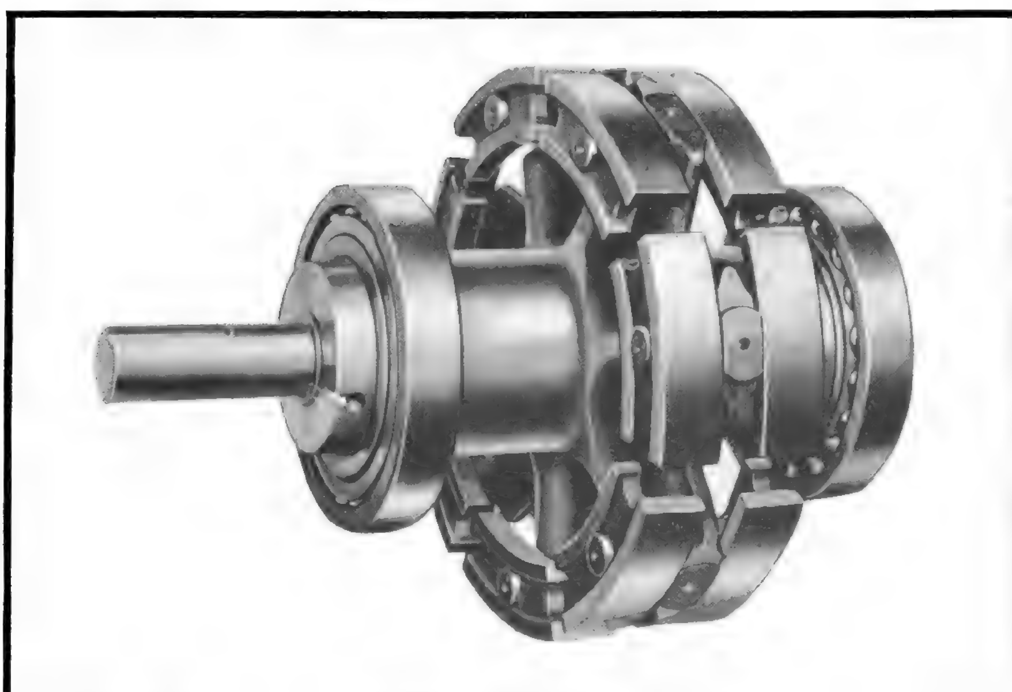

Figure 209

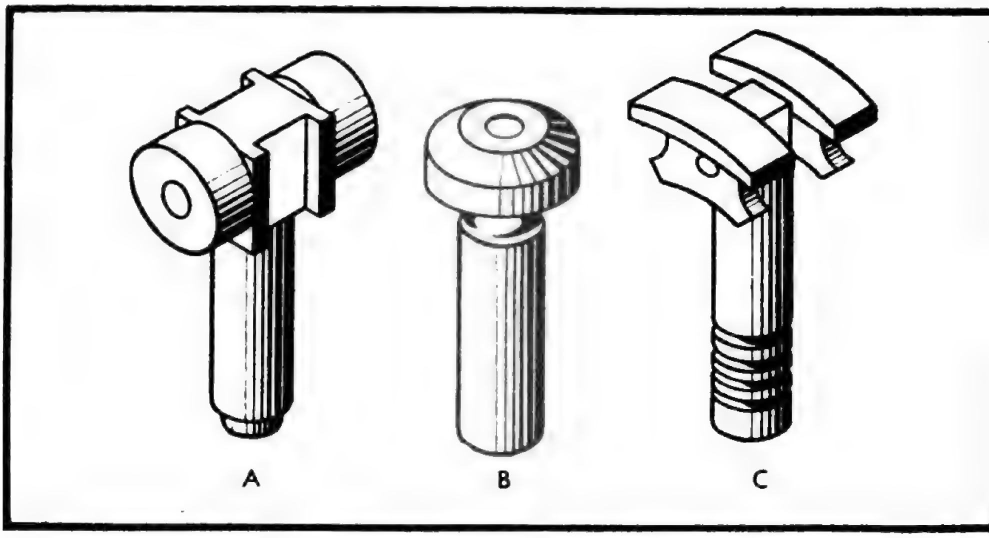

Figure 210

The pistons. Here too, different designs are used to serve the same general purpose. Some of these designs are shown in Figure 210. *A* shows a piston with small wheels that roll around the inside curve of the rotor. It is used in the Northern pump design. *B* shows a piston in which the conical edge of the top bears directly against the reaction ring of the rotor. It is used in Oilgear pumps. In this particular design while the piston is going in and out of the cylinder it will also rotate about its axis, so that the top surface will wear uniformly. *C* shows a piston attached to curved plates, as used in the Hele-Shaw pump. The curved plates bear against and slide around the inside surface of the rotor.

The sides of the pistons are accurately machined to fit the cylinders so there will be a minimum loss of liquid between the walls of the piston and cylinder. No provision is made for the use of piston rings to help seal against piston leakage.

The rotor. Here again the design may differ from pump to pump, as shown in Figure 211, but the rotor consists essentially of a circular ring, machine finished on the inside, against which the pistons bear. *A* is used in the Hele-Shaw pump; *B* in the Oilgear.

As we know, the rotor rotates within a slide block which can be shifted from side to side in order to control the length of stroke

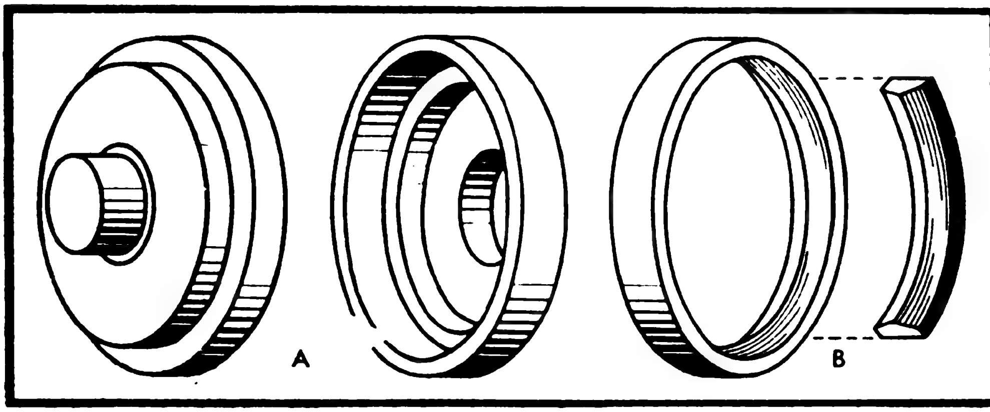

Figure 211

given to the pistons. The slide block has two pairs of machined surfaces on the exterior so that it can slide in tracks in the pump case. The sliding motion is usually controlled by a screw type fixture so that a close adjustment of the ring can be accomplished. Some pumps have an indicator which tells the amount the rotor is off-center in relation to the cylinder block, so that the operator will have an idea of the degree to which the pump's capacity is being utilized.

These parts, together with the drive shaft, constitute the main working parts of this type of pump. The drive shaft is connected to the cylinder block and is driven by an outside force such as an electric motor.

Oilgear fluid power pump. Figure 212 gives a simplified diagram of the operation of the Oilgear DR–6025 pump which is used in the operation of airplane catapult systems. This particular pump takes in liquid from the reservoir through the two upper holes bored through the shaft to the pintle, while it delivers liquid under pressure from the two lower holes. It can be adjusted, in other words, to deliver any quantity of liquid from zero to the maximum in one direction, but the direction of delivery cannot be reversed.

The slide block is held in position on one side by a spring, and on the other by liquid pressure coming from the control gear pump and applied against the control gear pump piston. A pilot valve plunger (not shown in the diagram) standing between the control

gear pump and the control gear pump piston is held in position electrically so that sufficient pressure is delivered to keep the slide block in position. When the pressure reaches a predetermined level, in order to avoid overload an electrical control is de-energized, liquid pressure is cut off from the control gear pump piston by the movement of the pilot valve plunger, and the adjustment spring of the slide block acts to center the slide block. Thus, the excess pressure condition is relieved by stopping the flow of liquid from the pump.

Two Oilgear pumps, each driven by its own electric motor and each

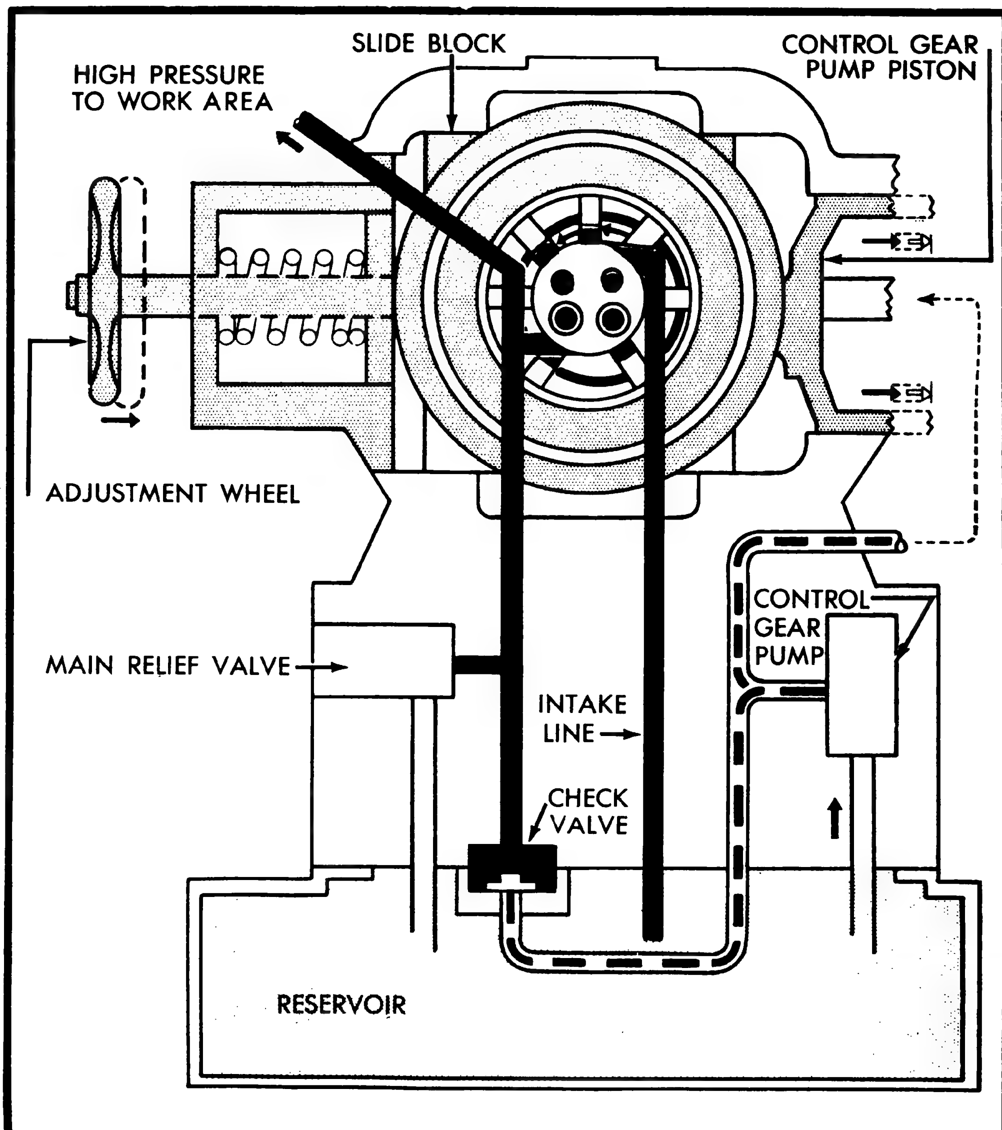

Figure 212

with its own supply of oil, can be used together to control the steering gear of a ship. Either pump can supply any quantity of oil desired, from zero to the maximum for the pumps used. Changes in the position of the steering wheel cause one or the other pump to deliver a supply of oil to the cylinders, controlling the position of the rudder. Movement of the steering gear control determines which pump shall be utilized, and to what degree. After the rudder has been shifted by hydraulic action, the steering gear control returns the pump control levers to neutral, so that the rudder is held in position without further pump action.

Oilgear fluid power pumps are made in a variety of sizes, capable of delivering from 3.7 to 104 gallons per minute at 1100 pounds per square inch pressure, 8.7 to 108 gallons at 1700 pounds pressure, and 1.7 to 108 gallons at 2500 pounds pressure. The smaller pumps work at 1140 revolutions per minute, and the larger pumps at 860 revolutions.

Northern radial piston pump, Series 5000. That this pump acts according to the same principle as the Oilgear pump will be evident by comparing Figure 213 with Figure 206. It differs only in minor details of construction. The pistons, for example, are of type *A* in Figure 210. The pump is used, like the Oilgear, for controlling steering gear, and likewise for airplane catapults, as well as for a variety of other purposes.

Northern radial piston pumps are constructed to deliver from 1.15 to 286 gallons per minute at 1000 pounds per square inch pressure, from 1.1 to 272 gallons at 2000 psi, and from 1.0 to 258 gallons at 3000 psi. Speeds of revolution vary from 400 per minute for the pumps of larger capacity to 1150 for those of smaller capacity, and the pumps weigh from 47 to 7100 pounds.

Hele-Shaw pump. This pump is manufactured by the American Engineering Company; Figure 214 shows the pump in an exploded view. All the parts shown fit inside the pump casing, except the two flanges, which are bolted one on each side of the casing. The piston used is shown in *C* of Figure 210. Each of the slippers at the head of

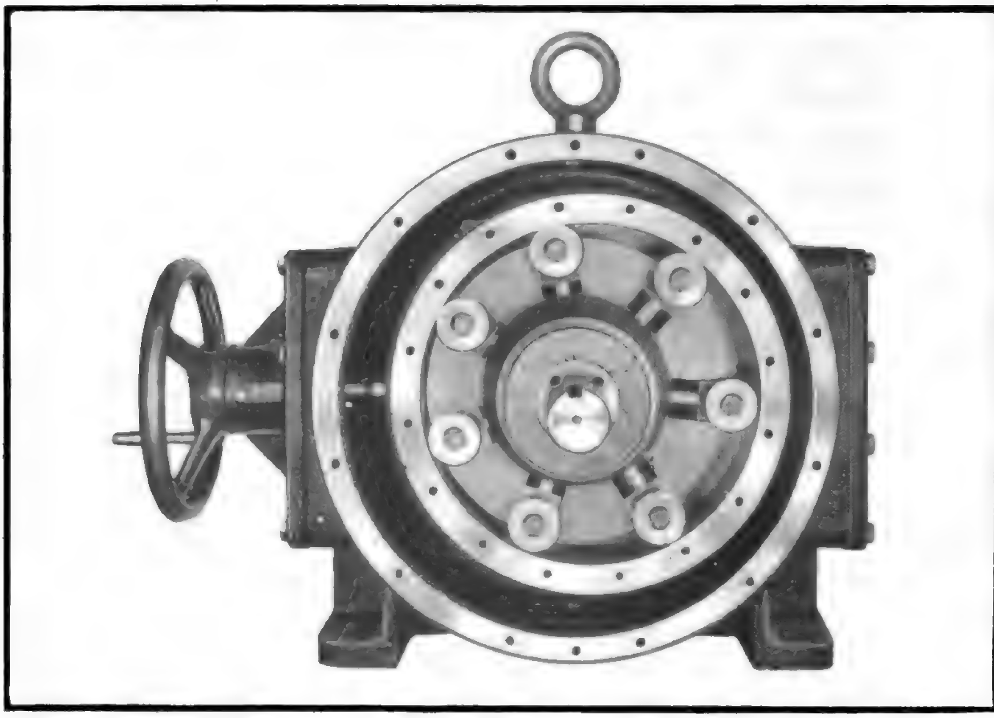

Figure 213

the pistons fits into a groove in the rotor surrounding the cylinder body (*A* in Figure 211). It is the changes in the position of this rotor or floating ring that controls the volume and direction of discharge of the pump. There should be no difficulty in seeing that this pump works in the same general manner as the other radial piston pumps already described.

Hele-Shaw low pressure pumps give from 7.5 to 150 gallons per

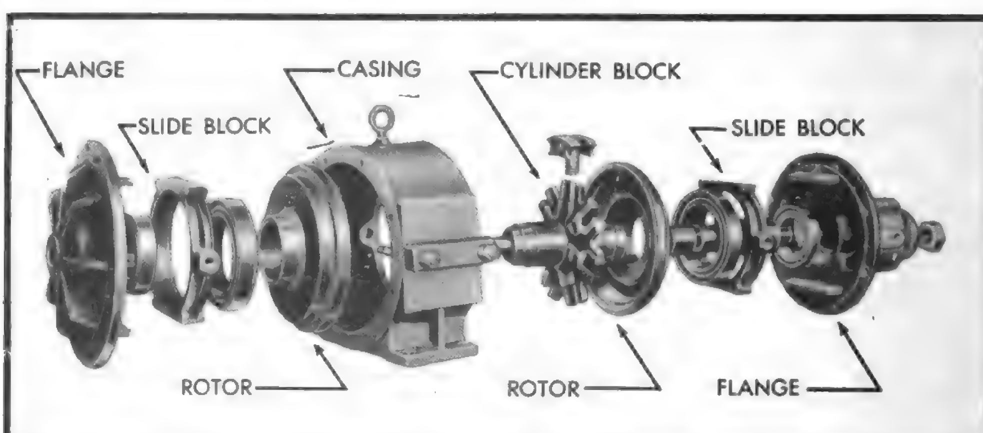

Figure 214

minute at from 1150 to 580 revolutions per minute at a maximum pressure of 1200 pounds per square inch; while the high-pressure pumps give from 3.75 to 120 gallons per minute at from 1150 to 580 revolutions per minute at maximum pressures of from 2000 to 3000 pounds per square inch.

Radial Piston Motors

The radial piston pump is sometimes used to operate a hydraulic ram, where straight-line motion is desired, but it is perhaps more frequently used in connection with a radial piston motor. This motor is really a pump used in reverse, since it is driven by liquid supplied by another pump, rather than itself pumping liquid.

The pump controls the motor in such a way that it is possible to use the combination as a speed reducer. This kind of speed reducer has many advantages over the mechanical type. The primary advantage is that the hydraulic reducer permits speed reduction of any amount from maximum pump speed to zero while keeping the source of power running at a constant speed. All mechanical reducers are definitely limited in the degree to which the speed can be reduced and few if any mechanical reducers permit the large range that the hydraulic reducer allows.

The radial piston motor is almost identical to the radial piston pump. The primary difference is that the motor has a fixed rotor which cannot be adjusted as was possible in the pump. To control the motor speed it is necessary to control the output capacity of the pump. This is done by controlling the degree to which the slide block of the pump is off-center in relation to the center of the cylinder block. Pressure in the system is practically independent of the volume output of the pump, and thus the speed of the motor, but varies with the load on the output shaft of the motor. In the radial piston pumps already studied we found that, as the cylinder block revolved, the pistons pressed against the rotor and were forced in and out of the cylinders, thereby receiving liquid and pushing it out under pressure. In the motor we have just the reverse motion—liquid pushed into the cylinders under pressure

drives the pistons outward. The piston pushing against the rotor causes the cylinder block to revolve.

Before going into too much detail about the motor we should find out why pushing the piston out will cause the cylinder block to turn. It will be necessary to call to mind a few simple facts in order to explain this action.

We know from previous experience that if we introduce liquid under pressure into a cylinder the piston will be pushed outward as shown in Figure 215. Let us suppose we have a wheel or rotor with a cylinder block inside which contains, say three pistons. The cylinder block is set off center in relation to the rotor, as shown in Figure 216.

If we introduce liquid into cylinder *1*, the piston must move outward since the liquid cannot be compressed and two bodies cannot occupy the same space. In this case, for the piston to be able to move out the cylinder block must revolve in a clockwise direction, since the piston in moving out will seek the point of greatest distance between the cylinder block and the rotor. As the force acting on piston *1* causes the cylinder block to move, piston *2* will start to change position and will approach the position of piston *3*. It should be noted that the distance between the cylinder block and the reaction ring of the rotor gets progressively shorter on the top half of the rotor as it is shown in Figure 216. As piston *2* moves

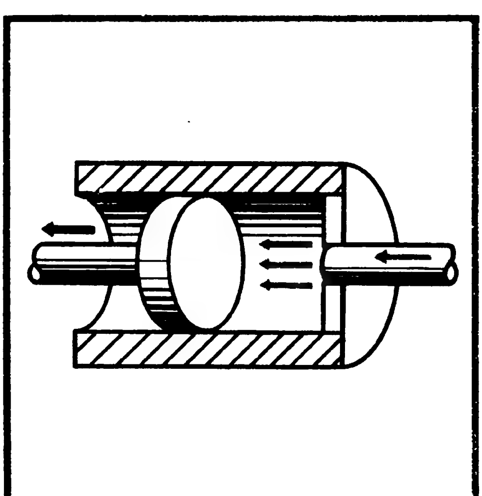

Figure 215

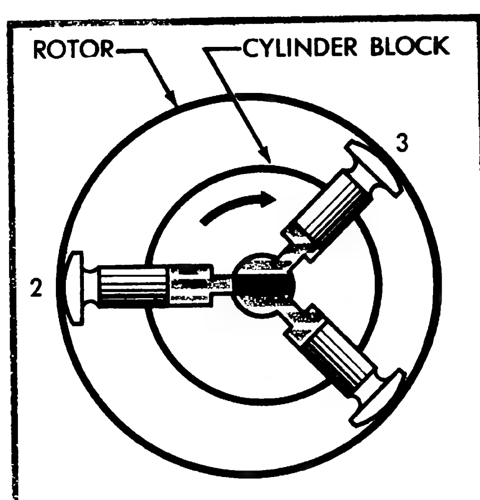

Figure 216

it will be pushed inward and will force the liquid out of the cylinder. Since there is little or no pressure on this side of the pintle valve, the piston is easily moved in by contact with the reaction ring of the rotor, and the liquid is easily pushed out of the cylinder and back to a reservoir or back to the inlet side of the pump. As piston *3* moves past the midpoint, or past the shortest distance between the cylinder block and rotor, it comes to the pressure side of the pintle valve and liquid is forced into the cylinder. Piston *3* then becomes the pushing piston and in turn rotates the cylinder block. This action continues as long as liquid enters the pistons under pressure to push the pistons outward and permit the discharge of liquid under a smaller pressure.

The rate at which we pump liquid into the cylinders determines the speed at which the cylinder block will turn. We found in the pump of this type that we could control the amount of flow by controlling the length of the piston stroke. The length of stroke was controlled by the amount the cylinder block is off-center with respect to the reaction ring of the rotor.

Suppose in a pump-motor combination of this type in which both units are the same size, the rotor is set off-center in the pump a distance that is just equal to the offset of the rotor in the motor end. For each full discharge of one cylinder in the pump a piston in the motor must move an equal distance, as it will receive the same amount of liquid that the pump discharges. The cylinder block of the motor end will revolve at the same speed as the cylinder block of the pump end. Now suppose the offset distance in the pump is shortened, so that it takes the discharge of two cylinders in the pump end to fill one cylinder in the motor end. Under this condition the cylinder block of the pump will revolve twice for each revolution of the cylinder block in the motor; thus the motor end will run at just half the speed of the pump end.

Since we have a wide control over the pump end, we have an equal control over the motor end—a control achieved by regulating the amount of flow from the pump. If, in the pump, we exactly center the rotor and the cylinder block, no pumping action will take place,

consequently no liquid will be delivered to the motor end, and therefore the output shaft of the motor will not rotate.

In the pump, the direction of flow is reversed by moving the rotor from one side of the neutral position to the other. In the motor, the position of the rotor is fixed. Action of the motor is reversed, however, by reversing the output of the pump. Thus, liquid enters the motor on the top side of the pintle valve, which causes piston *3* in Figure 216 to become the piston into which liquid enters under pressure. This will cause the cylinder block to revolve in a counter-clockwise direction.

The parts of the motor are identical with those of the pump, with the exception of the control device. The two machines differ only in that the rotor in the motor is fixed and cannot be shifted from side to side as it can be in the pump.

The pump end of this unit is often referred to as the A-end and the motor end as the B-end. The pump and motor (A-end and B-end) may be housed in a single housing (C type), or the pump and motor can be widely separated and connected by appropriate piping (K type).

The radial piston pump-motor combination finds its widest usage as a hydraulic drive in connection with steering gears, anchor windlasses, and capstans. Because of the degree of leakage involved, and the consequent lack of precision in control due to varying losses, this pump cannot be used for turret training purposes.

Lubrication of the working parts and the rotor is obtained through oil leakage around the pistons. Leakage oil is thrown out by centrifugal force to lubricate the floating ring, piston slippers, and all bearings. Excess oil accumulates in the casing and is discharged to a sump tank from which it is returned to the replenishing tank by a separate pump. As lubrication is furnished by leakage oil, extreme care should be exercised not to operate this pump in the neutral or no stroke position more than necessary as, in this position, no lubrication is provided

Installation, Maintenance and Casualties

The best and most specific source of information on these subjects for each pump is the instruction pamphlet prepared by the manufacturer. It tells in detail what operations the pump can perform, how it is constructed, how it should be installed, operated and disassembled, the proper oils or lubricants to use, how to avoid trouble, what troubles are likely to occur, and how they can be corrected. Information of this kind is usually rather detailed, and may apply only to a particular make of pump or even to a particular model. The instructions presented are a result of experience, and should be followed except where a sound reason can be given for departing from them.

The general considerations already discussed in Chapter 8, to be found in greater detail in Chapter 47 of the *Bureau of Ships Manual*, should also be familiar to anyone responsible for the operation of these pumps. Here as elsewhere in hydraulic systems foreign matter can work havoc. Cylinders and pistons in these pumps are particularly subject to wear. Symptoms of trouble should be investigated immediately, before the condition becomes aggravated. Among these symptoms are failure to function, sluggish control, inability to control, insufficient volume, excessive heating, noise, and undue losses through leakage.

QUESTIONS

1. What is variable delivery, and how is it obtained?

2. How does the characteristic action of valve controlled pumps differ from that of variable delivery pumps?

3. Where and how does liquid enter and leave the pump chamber of a radial piston pump?

4. How is flow reversed in a radial piston pump?

5. Explain the following parts of a radial piston pump, and show in general how they are related: pintle, cylinder block, pistons, rotor with its reaction ring, source of power. Which parts rotate, and which remain stationary?

6. Explain in a general way how a variable delivery pump can be used to steer a ship.

7. How can a pump be used as a motor? Why is this done? What is meant by the *A*-end and the *B*-end?

8. How is the speed of rotation of a hydraulic motor related to the speed of rotation of its pump? To the speed of rotation of the source of power?

9. How is the direction of rotation of a hydraulic motor reversed?

10. State some of the casualties likely to occur with radial piston pumps.

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Chapter 12

AXIAL PISTON PUMPS AND MOTORS

In this chapter we explain the principle on which axial piston pumps are constructed, including the universal joints basic to their operation. We then proceed to a discussion of the ways in which these pumps are used as hydraulic motors. Three varieties of axial piston pumps are described—the Waterbury, the Northern and the Vickers.

Principle of the Axial Piston Pump

In axial piston pumps, a cylinder block with its pistons is rotated on a shaft in such a way that the pistons are driven back and forth in their cylinders in a direction parallel to the shaft. This is called axial motion. Liquid is forced into the cylinders by pressure differences while the pistons are moving outwards, and is driven out by the pistons while they are moving inwards.

Williams universal joint. This pumping action is made possible by a universal joint. To explain how the joint works, we shall begin by constructing step by step one of the Williams, Hooke or Cardan type. This is the kind used in many of these pumps.

Let us install a rocker arm on a horizontal shaft (Figure 217). The arm is joined to the shaft by a pin in such a way that it can be swung back and forth. Next we place a ring around the shaft

and pin it to the rocker arm so that the ring can turn from left to right as shown in Figure 218. We can now get two rotary motions in different planes at the same time, and in varying proportions as may be desired. The rocker arm can swing back and forth in one arc, and the ring can simultaneously move from left to right in another arc, in a plane at right angles to the plane in which the rocker arm moves.

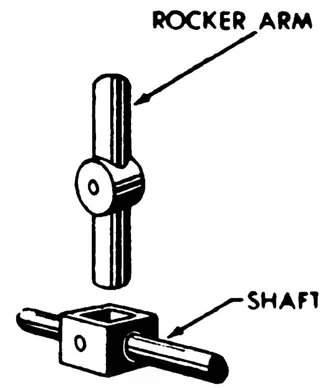

Figure 217

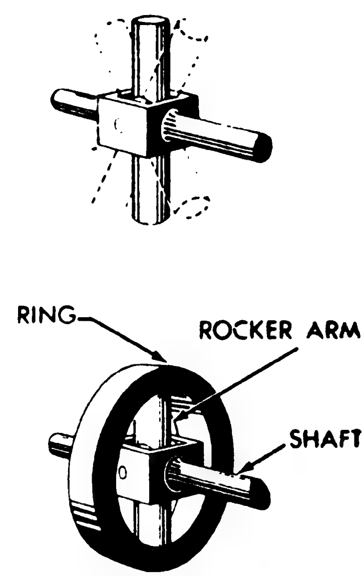

Figure 218

Now let us add a tilting plate to our assembly (Figure 219). For the time being let the tilting plate stand at a slant to the axis of the shaft. In this position the rocker arm is slanted back at the same angle as the tilting plate, so that it lies parallel to it. The ring is also parallel to the tilting plate and in contact with it. Its position in relation to the rocker arm is unchanged from that shown in Figure 218.

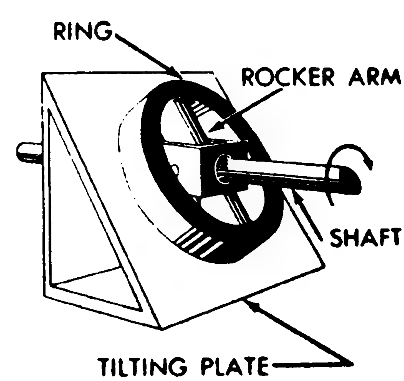

Figure 219

Now let us keep the shaft horizontal and rotate it in a clockwise direction through a quarter of a turn, to the position shown in Figure 220. The rocker arm is still in the same plane as the tilting plate, and is now perpendicular to the axis of the shaft. The ring has turned on the rocker arm pins, so that it has changed its position in relation to the

rocker arm, but it remains parallel to, and in contact with, the tilting plate.

Now let us rotate the shaft another quarter turn, as shown in Figure 221. The parts are now in the position shown in Figure 219, but with the ends of the rocker arm reversed. The ring still bears against the tilting plate.

If we continue to rotate the shaft, the rocker arm and the ring will turn about their pivots, with each changing its relation to the other, and with the ring always bearing against the plate. This will be so even if we change, within limits, the angle of the plate.

Now let us go a step farther and place a wheel upright on the shaft, as shown in Figure 222. The wheel is fixed to the shaft, so that it rotates with the shaft. Let us also add two rods *A* and *B*, loosely connected to the tilting ring, and running through two holes standing opposite to each other in the fixed wheel. As the shaft is rotated, the fixed wheel will run perpendicular to the shaft at all times. The tilting ring will rotate with the shaft and will always remain tilted, since it will remain in contact with the tilting plate. We can see from the diagram that the distance along shaft

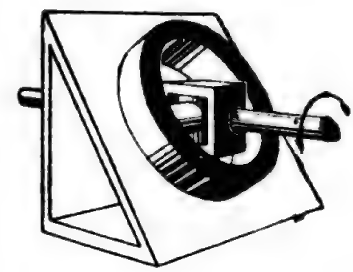

Figure 220

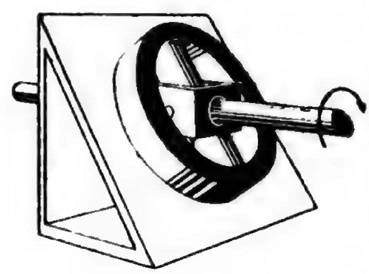

Figure 221

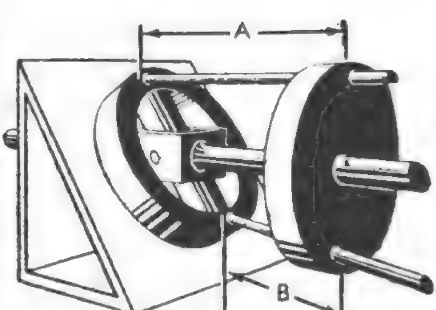

Figure 222

A, from the tilting ring to the fixed wheel, is greater than the distance along shaft *B*. As we rotate the assembly, however, the distance along shaft *A* will decrease as its point of attachment to the tilting ring comes closer to the fixed wheel, while the distance along shaft *B* will increase. These changes will continue until after a half revolution the initial position of the two rods has been reversed. After another half revolution the two rods will again be in their original positions.

From this we can see that the rods are moved in and out through the holes in the fixed wheel as the assembly rotates. This is the way the axial piston pump works. To get a pumping action, it is only necessary to place pistons at the ends of the rods, beyond the fixed wheel, and insert them in cylinders. The rods must be connected to the pistons and to the wheel by ball and socket joints. As the assembly rotates, each piston will move back and forth in its cylinder. Intake and discharge pipes can be arranged so that liquid will enter the cylinders while the spaces between the piston heads and the bases of the cylinders are increasing, and will leave them during the other half of each revolution, when the piston heads are moving in the other direction.

How the joint gives variable displacement. Reference to Figure 222 will make it clear that the distance the pistons will move back and forth in their cylinders will depend on the tilt given to the tilting box. With no tilt at all, no pumping action would occur, since the pistons would not move back and forth at all. The distances *A* and *B* in Figure 222 would equal, and would remain equal as the assembly rotates. If the angle of tilt given to the tilting box were reversed, making distance *A* less than distance *B*, the pumping action would be reversed. What had been the discharge would now become the intake, and vice versa. By adding a mechanism to control the angle at which the tilting box stands, therefore, any variation in delivery can be obtained, from a maximum in one direction through zero delivery, to a maximum in the opposite direction, although the drive shaft continues to rotate at a constant speed.

Lag and surge periods in the Williams joint. The Williams joint

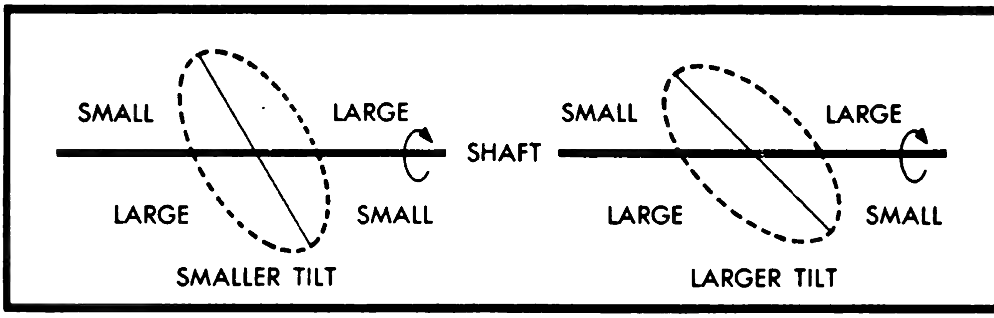

Figure 223

we have just described was the first to be used, in the Waterbury pump, to give variable delivery in an axial piston pump. While many new possibilities were opened up by this development, the Williams joint suffers from a mechanical irregularity in its operation.

As the rocker arm and ring rotate at an angle to the driven shaft, they approach and move away from the axis of the shaft twice during each revolution. As shown in Figure 223, they will swing over a relatively smaller arc in relation to the shaft in two opposite quadrants, and over a relatively larger arc in the other two quadrants. The difference between the arcs traversed increases as the tilt is increased. When the rocker arm and tilting ring are relatively close to the axis of the drive shaft, they will travel over a relatively small distance for a given amount of rotation of the drive shaft. When they are relatively far away, they will travel over a relatively greater distance for the same amount of rotation of the shaft.

Turning now to the pistons moving back and forth in their cylinders, the result of this irregularity in rotation is that the pistons will move more slowly in one pair of quadrants, depending on the tilt, and more quickly in the other pair. Delivery of liquid will therefore lag behind during one pair of quadrants, and surge ahead during the other pair. The lags occur when the rocker arm and tilting ring must sweep through an angle of less than 90° while the drive shaft is rotating 90°. The surges take place when the arm and ring must sweep through more than 90° while the drive shaft is rotating 90°.

The delivery of liquid by the pump will fluctuate, even though the average speed of revolution of the drive shaft and the driven mechanism is exactly the same. The amount of fluctuation will differ with the tilt given to the tilting box. For a tilt of 15°, the total variation above and below the drive speed will be 7 per cent, half on each side of the average speed. For a tilt of 30° it will be 29 per cent, and for a tilt of 40° it will be 54 per cent.

These fluctuations can be corrected to some extent by spacing the cylinders irregularly around the circumference of the cylinder block, as shown in Figure 224. For the nine-piston pump there shown, intake and delivery is accomplished from 2½ cylinders over each of the quadrants involved in a lag period, because the pistons move more slowly then. For the surge quadrants intake and delivery goes on from 2 cylinders for each quadrant, because the pistons move more quickly then. The added quantity of liquid handled in one pair of quadrants, and the decreased quantity handled in the other pair, help to compensate for the mechanical irregularity in the action of the joint.

The arrangement described corrects the irregularity for a 14.5° angle of tilt—the angle that was selected as most needing correction in view of the conditions under which these pumps are used on gun mounts. An alternative method of correction is to put the cylinders closer to the center of the cylinder block in lag quadrants, and farther from the center in surge quadrants. Then the driven part of the assembly will rotate faster to take the liquid served to it during the lag period, and more slowly during the surge period. This method is used on the B-end of the Waterbury pump-motor combination, where the tilting plate is permanently set at an angle of 20°, so that the other correction would not be adequate.

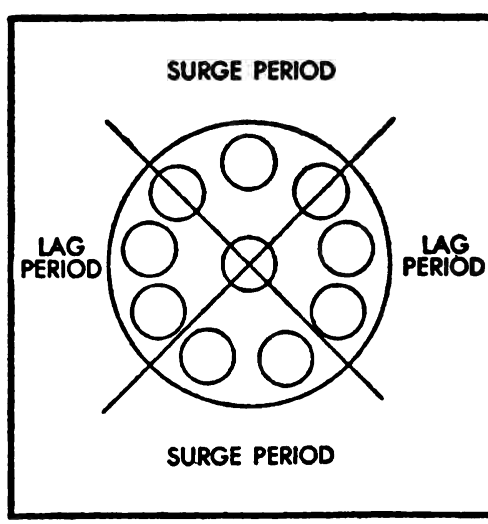

Figure 224

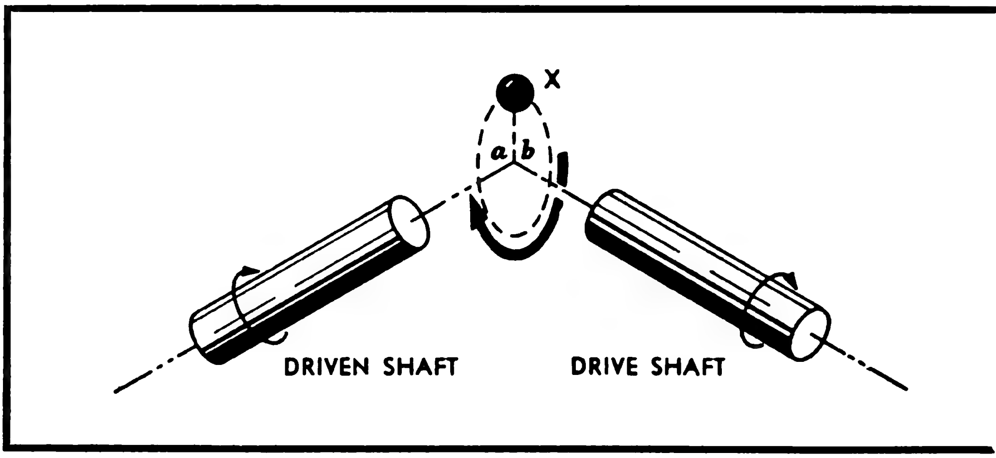

Figure 225

Another method of compensating for the irregularity is to use two Williams points set up in such a way that the irregular action of each joint compensates for that of the other. The driven shaft is connected to the drive by an intermediate shaft, which must be set to make the same angle with each of the other shafts. The drive and driven shafts must lie in the same plane. The Vickers axial piston pump makes use of two Williams universal jòints.

Bendix-Weiss constant velocity universal joint. A number of constant velocity universal joints have been developed, the one manufactured by Bendix Aviation is used in many Northern axial piston pumps. This joint is based upon a geometrical principle which must be understood in order to grasp how the joint works.

Figure 225 shows a drive shaft and a driven shaft standing at an angle to each other. Let us suppose that the drive shaft rotates the driven shaft through a pair of fingers set around a ball *X*. This ball carries the load of rotation in such a way that angle *a* is equal to angle *b*. As the drive shaft rotates in the direction shown, *X* will move in the same direction in a circular path, always keeping angle *a* equal to angle *b*. After a complete revolution of the shaft it will have returned to the original position. Since angle *a* has been kept equal to angle *b*, the driven shaft will rotate at exactly the speed of the drive shaft.

An arrangement which accomplishes this result is used in the Bendix-

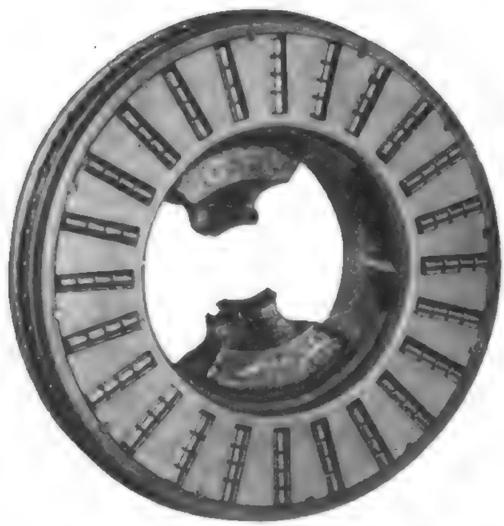

Figure 226

Figure 227

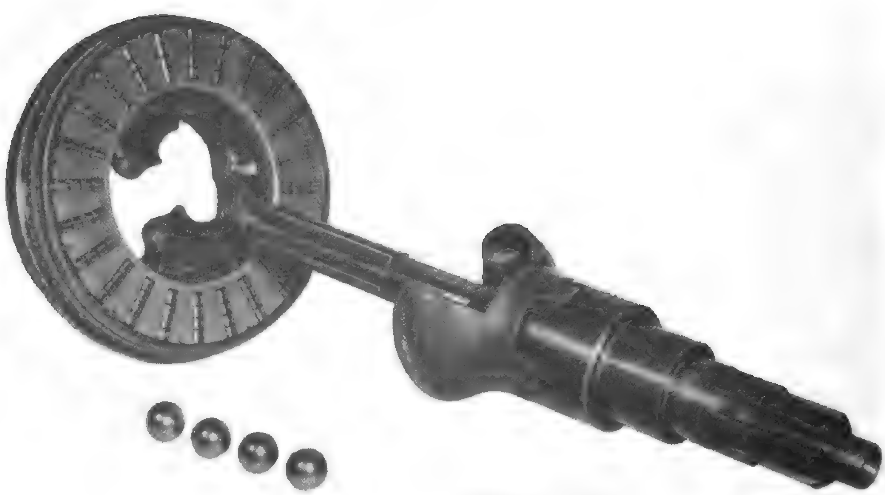

Figure 228

Weiss universal joint. The force of rotation is carried by a pair of balls which move back and forth in specially shaped runways or races as the drive shaft rotates. The races are given such a shape that the balls always remain in a single plane as they move back and forth in their races. Angle *a* will always equal angle *b* as the drive shaft rotates, just as in Figure 225, and the units connected by the joint will therefore rotate at the same speed.

In the actual Bendix-Weiss joint four balls are used. One pair of balls carries the force transmitted in clockwise rotation, and the second pair that transmitted in counterclockwise rotation. There are four races, one in each quadrant. One face of each pair of races is grooved in a jaw welded to the socket ring of the pump, as shown in Figure 226, while the other face is grooved in a jaw set on the drive shaft, as shown in Figure 227. The two jaws fit into each other with a ball held between each pair of races. An exploded view of the position of the drive shaft in relation to the socket ring for a certain angle of tilt is shown in Figure 228. The races move back and forth slightly as the two members rotate, and always take the position described above, so that rotation of the driving member is exactly transmitted to the driven member.

The Navy uses the Bendix-Weiss joint on many single gun mounts. It is a component of many Northern axial piston pumps.

Waterbury Variable Speed Transmission

The pump (A-end). In this pump, shown in an exploded view on the right side of Figure 229, which gives a complete pump-motor combination, our fixed wheel becomes a cylinder barrel, the tilting ring becomes a socket ring, the tilting plate becomes a tilting box, and the rods between the tilting ring and the fixed wheel become piston rods with pistons attached to them. The pistons are pushed back and forth in their cylinders as the shaft is rotated.

Other parts shown are the valve plate, tilting box control and drive shaft. The tilting box control provides a means of changing the angle of the tilting box to the drive shaft, thereby controlling the length of the piston strokes. When the tilting box is at right angles to the shaft

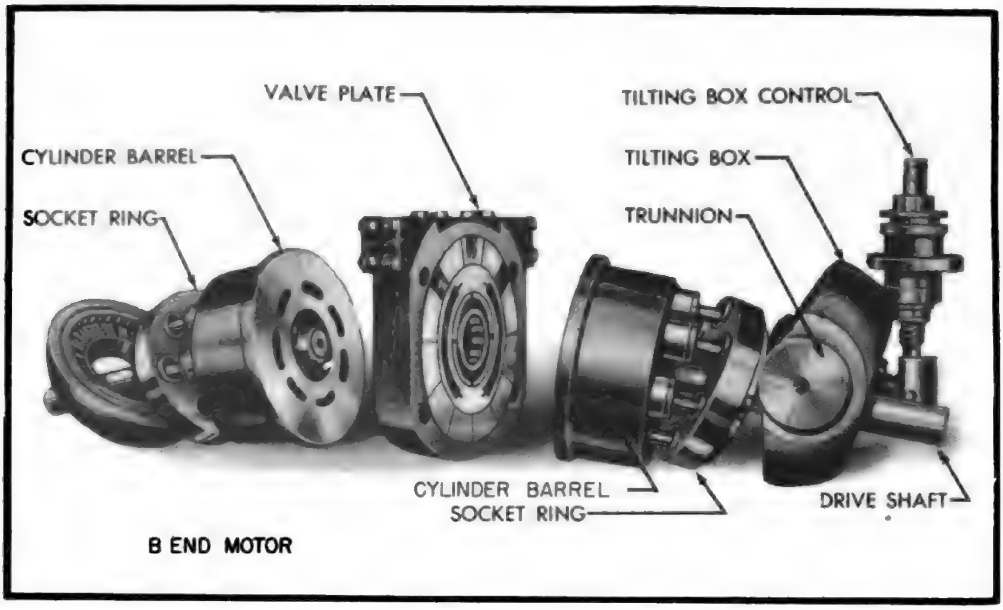

Figure 229

it is parallel to the cylinder barrel, and no pumping action will take place even though the shaft is rotated, since the pistons will not move back and forth in their cylinders. As the tilt on the tilting box is increased, the piston strokes become longer, and more liquid is pumped with each revolution of the shaft.

Parts of the pump. The valve plate (Figure 230) has a carefully machined surface against which the face of the cylinder barrel slides as it rotates. Through the plate are two sausage-shaped passages called valve-plate ports, one in each half of the plate, through which the liquid passes from the intake or to the discharge. Between the ports at the top and bottom are flat faces called lands, into which are cut small extensions from the ports. As the cylinder barrel rotates, the cylinder ports pass across these lands and the contents of each cylinder is imprisoned within the cylinder. The face of the valve plate has grooves cut into it which act as oil seals to interrupt leakage, and also to trap dirt. Figure 229 shows these grooves as they appear on later models, while Figure 230 shows them as they appear on an earlier model.

A small amount of oil necessarily leaks from the high-pressure active part of the pump into the low-pressure inactive body of oil in the

pump case. If this leakage is not replaced as fast as it occurs, a low-pressure condition will be built up in the cylinder and port passages. Hence there are two check valves in the lower part of the valve plate called replenishing valves (only one shown in Figure 230). One of these is connected with each passage, and permits the oil to flow freely from the casing space into the port passage, but prevents it from flowing in the opposite direction. The valve itself is a steel piece screwed into the pump from the outside.

With low speed transmission of power, the oil pressure may rise to thousands of pounds per square inch if the resistance to be overcome is correspondingly great. It is therefore necessary to provide relief valves set at the desired maximum pressure. When pressure exceeds this amount, the oil escapes from the high-pressure port passage through the relief valve into the casing space and flows back again through the replenishing valve into the low-pressure port passage. In more recent models of the Waterbury pump, the replenishing valves are placed inside the relief valves. The action of one is independent of the action of the other, since the relief valves act under high-pressure conditions and the replenishing valves under low.

At the highest points in the two port passages are needle air valves,

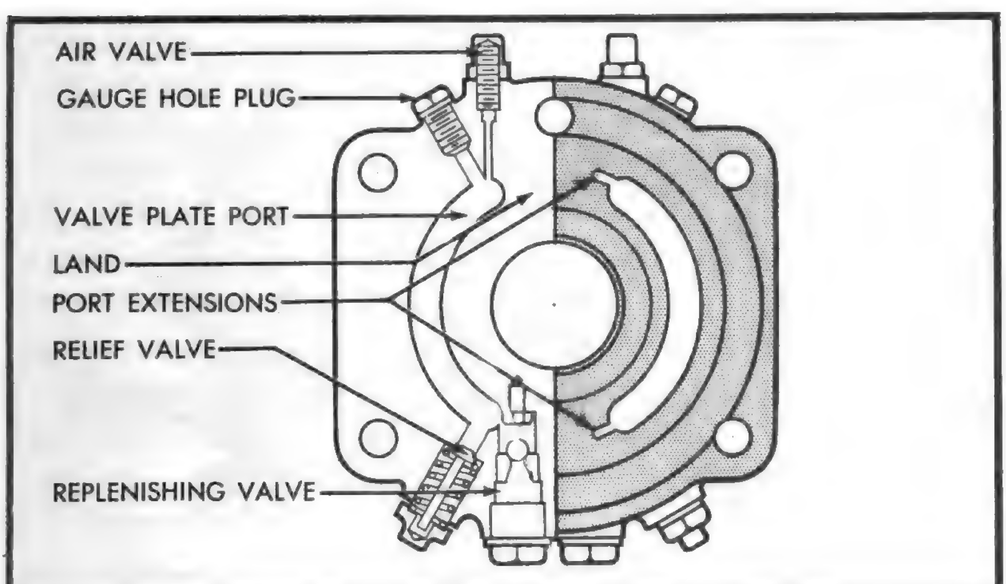

Figure 230

which allow air imprisoned in the passages to escape into the casing space, where it can rise through an oil expansion box. These valves are opened one or two turns during the filling process, after which they are closed tight until the oil is once more changed. The valves are needed because pulsations in the pump-motor combination, whose operation will shortly be described, produce a suction that strongly tends to draw air into the system through any available joint or opening.

The cylinder barrel is shown in Figure 231. The number of cylinders is always odd. With an even number of cylinders, for example eight, at certain times two cylinders would be blocked by the valve-plate lands while three would be receiving and three discharging liquid. At other times, with no cylinders blocked by the lands, four would be receiving and four discharging liquid, or one-third more than previously. As a consequence, flow would be pulsatory. With an odd number of cylinders, say nine, four would always be receiving and four discharging.

The cylinder barrel is loosely keyed to the shaft in an endwise direction so that its face rests squarely against the face of the valve plate.

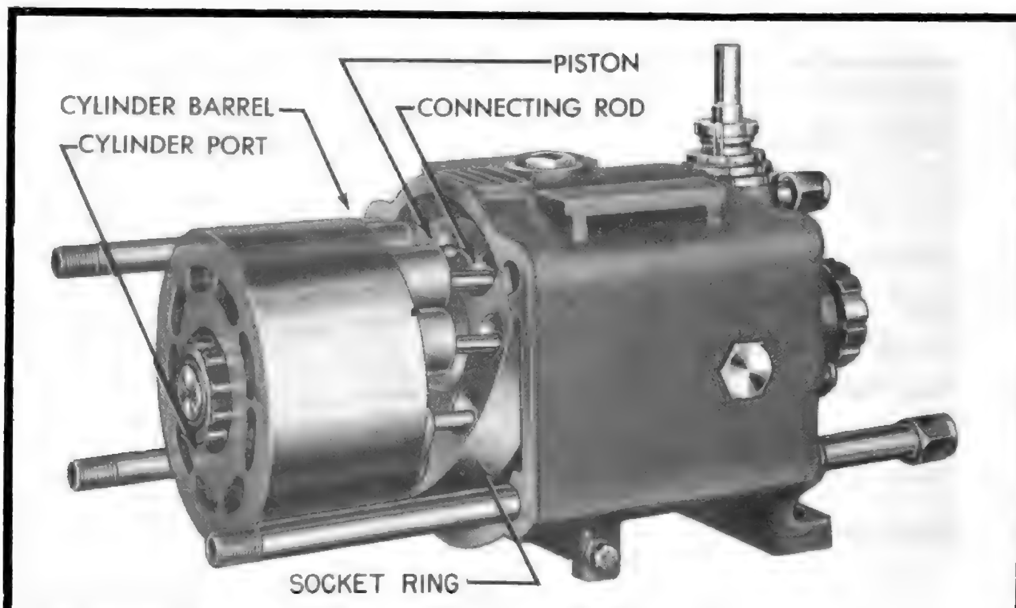

Figure 231

Endwise motion of the barrel is resisted by a barrel spring backing against a flange on the shaft. The purpose of the spring is to hold the barrel against the valve plate when the pump is not in operation. When the oil is under pressure the barrel is automatically held against the valve plate because the cylinder ports are smaller than the pistons, giving an unbalanced internal force which holds the barrel against the valve plate.

The pistons and cylinders are ground and lapped to a smooth working fit so that no packing is necessary. Narrow shallow grooves are cut around the pistons to interrupt leakage and trap dirt, and to give the pistons hydraulic balance. The face of each piston has a small hole bored through it which connects to the piston socket bearing and serves to lubricate the bearing. The open end of the piston is internally threaded to receive the socket bearing and the split ring bushing rod. These and other details of construction are shown in Figures 234 and 235, which show the Northern axial piston pump and motor. This pump is in many ways similar to the Waterbury pump. The open end of the piston is internally threaded to receive the socket bearing and the split ring bushing which completes the bearing over the ball end of the piston rod.

Connecting rods join the pistons to the socket ring. The rods have spherical ball ends which are the ball part of a ball and socket joint, permitting the pistons to move freely in their cylinders. Through the whole length of the connecting rod is a small hole which feeds oil under pressure from the hole in the piston through the connecting rod to the ball and socket joint in the socket ring.

The socket ring (Figure 232) has sockets cut into it which are fitted with bronze ring sockets against which rest the ball ends of the connecting rods. The back of the socket ring is provided with a roller track which has two roller faces, one for the main conical thrust rollers and the other for the diagonal thrust or cylindrical rollers (see Figure 226).

The body of the socket ring extends inwards and is secured to the universal joint parts. The action of the universal joint has already

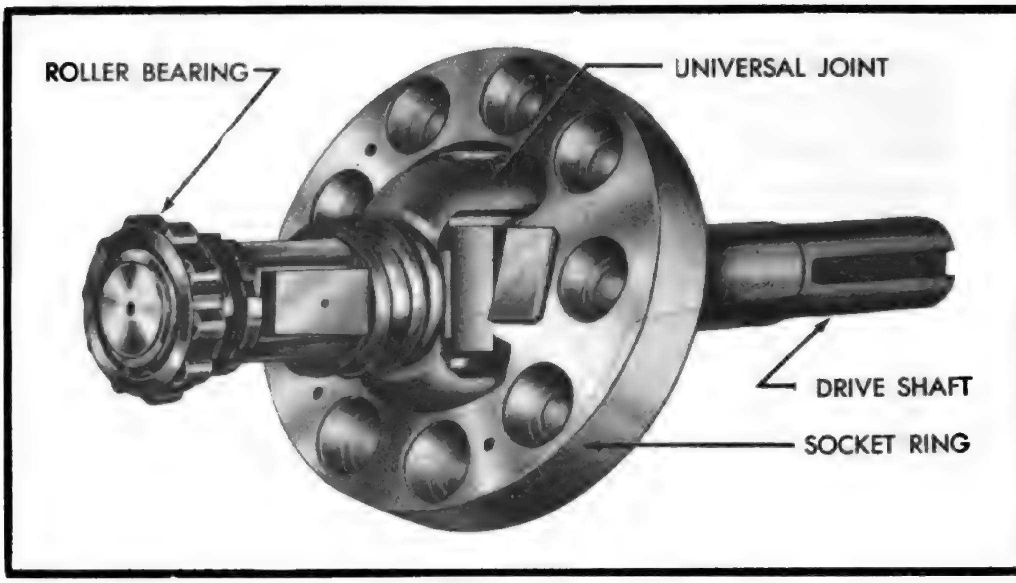

Figure 232

been explained. Figure 232 shows the socket ring connected to the shaft through the universal joint.

The purpose of the tilting box (Figure 229) is to carry a thrust roller track against which the socket ring can rotate in a plane at any desired angle to the shaft. The tilting box is suspended and can be oscillated on two trunnions, which are formed on the box itself, and which bear in bushings set in the sides of the case. An elongated hole is cut through the bottom of the box so as to give a free passage for the main shaft when the box is tilted to its maximum angle, which in most cases will be 20° from the vertical. Projecting from the bottom of the box are prongs or fingers to connect it to the control shaft. The tilting box contains roller bearings upon which the socket ring bears.

The tilting box control shaft, shown in Figure 229, is used to tilt the tilting box on its trunnions either way from the neutral or perpendicular position according to the direction and speed of liquid flow required. The control shaft is threaded and provided with an assembly which connects with the tilting box. Turning the control shaft causes the tilting box to be tilted at any angle to obtain the pumping action desired.

The drive shaft (Figure 229) is the shaft by which the pump is

activated. It has a roller bearing installed on the end which fits into the valve plate (Figure 232). The universal joint which holds the socket ring is incorporated into the yoke of the shaft. The shaft extends outside the housing and has the driving unit attached to it either directly or by means of belting, gears, or a chain drive.

The motor (B-end). The pump or A-end is frequently used in connection with a hydraulic motor or B-end. The A-end delivers liquid to the B-end which is used as a motor to perform some work operation by means of the liquid delivered to it by the pump. The only difference between the pump and the motor is that the tilting box of the motor is permanently set at an angle which amounts to 20° in some ordnance set-ups.

The hydraulic motor or B-end can be directly connected hydraulically to the pump, so that both the pump and the motor use the same plate (type C installation), or the motor can be set up at a distance from the pump, with the two mechanisms connected by piping (type K installation). A complete type C pump-motor unit is shown disassembled in Figure 229. A type K installation is shown in Figure 233.

The motor end of an axial piston pump works for exactly the same reason as the motor end of a radial piston pump, as explained in Chapter 11. Liquid introduced under pressure to a cylinder causes the piston to be pushed out. In being pushed out the piston through its connecting rod will seek the point of greatest distance between the cylinder barrel and the socket ring. The resultant pressure of the piston against the socket ring will cause the cylinder barrel and the socket ring to rotate. This action occurs during the half revolution while the piston is passing the intake port of the motor, which is connected to the pressure port of the pump. After the piston of the motor has taken all the liquid it can from the pump, it passes the valve plate land and starts to discharge the oil through the outlet ports of the motor to the suction inlet of the pump, and therefore to the suction pistons of the pump. The pump is constantly putting pressure on one side of the motor while it is constantly receiving liquid from the other side. The liquid is merely circulated from pump to motor and back again.

The speed of rotation of this unit is controlled in the same way as with the radial piston unit. When for each full stroke of a pump piston a motor piston must move an equal distance, speed of output will equal speed of input. If we change the tilt on the tilting box of the pump so that the piston stroke of the pump is only half as long as the stroke of the motor, it will take the discharge from two pumping pistons to fill one motor piston. The motor will then run just half as fast as the pump. If there is no tilt at all on the tilting box the pumping pistons will not move axially, and no liquid will be delivered to the motor side. The motor side will therefore deliver no power.

If we reverse the tilt on the tilting box of the pump, the direction of the flow will be reversed. Liquid will enter the motor through the port by which it formerly was discharged. This will reverse the direction of rotation of the motor. The motor rotates in the same direction as the pump when the two tilting boxes are tilted in the same direction, and in the opposite direction when the two are tilted oppositely.

With the pump that has been described, full control of the pumping action is possible, all the way from zero delivery to full delivery

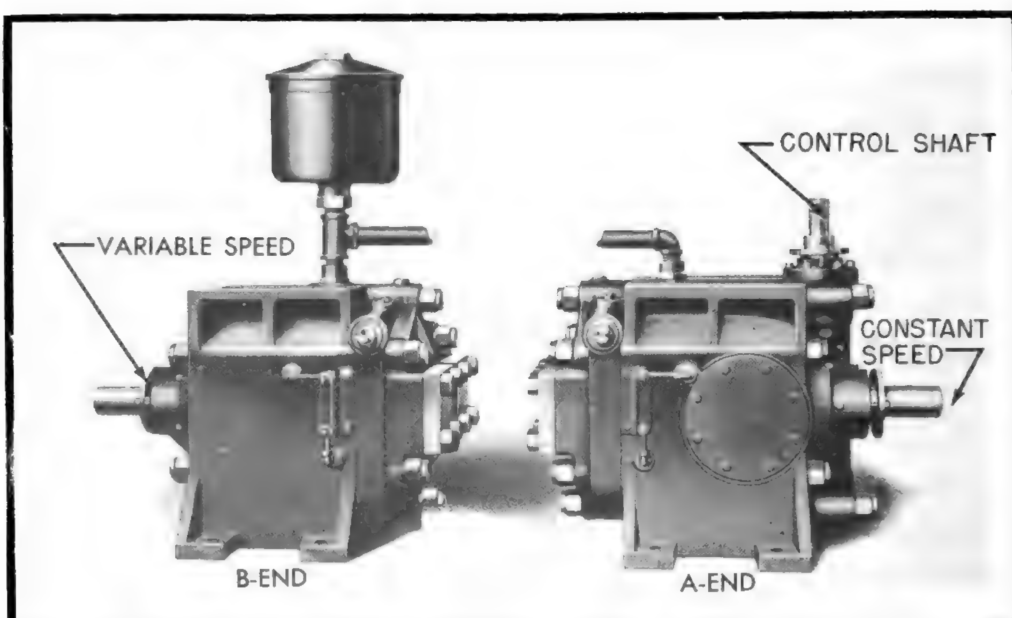

Figure 233

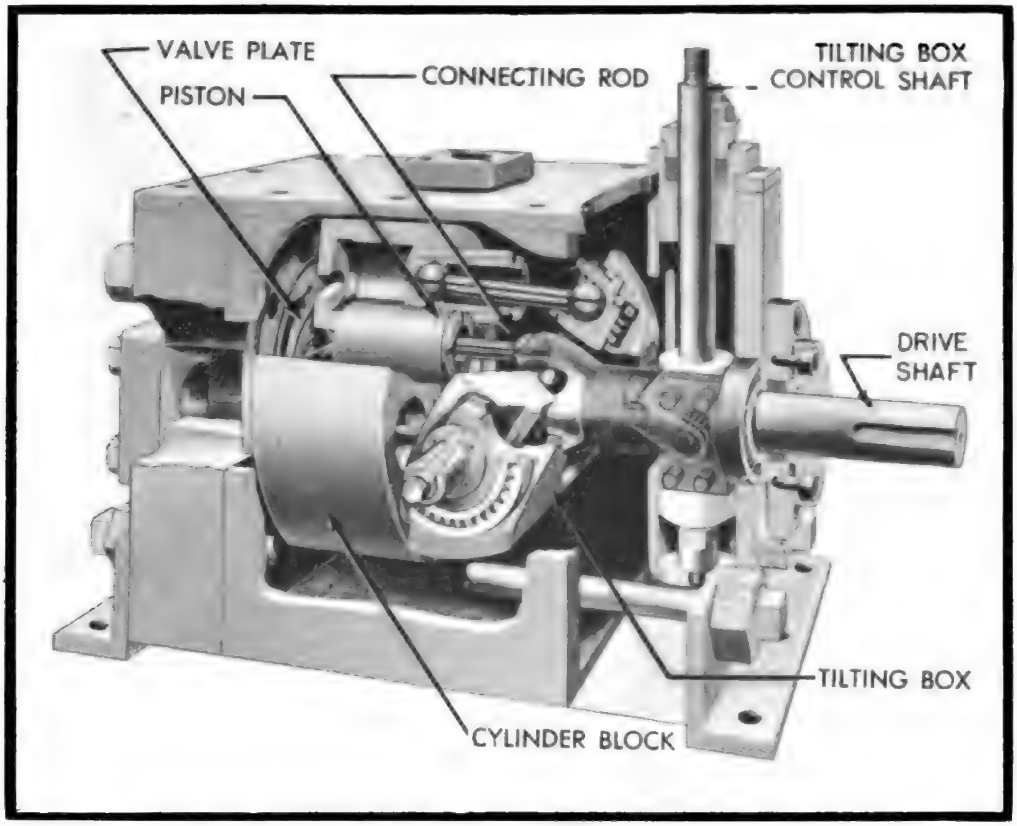

Figure 234

at the pump's maximum capacity, while the pump is always being operated at a constant speed. The volume of liquid delivered and the pressure set up in the system are independent of the speed of operation of the pump. Within the capacity of the pump, pressure depends entirely on load, so that any resistance whatever will be overcome up to that limit. If the load remains constant, the rate of volume output has no effect on the pressure. Under these conditions, the power required to run the pump would increase directly with volume output.

The Waterbury pump-motor is made in a half-dozen sizes. The revolutions per minute vary from 600 for the smallest size to 250 for the largest. Horsepower varies from 3 to 37.5 for a pressure of 100 pounds per square inch, up to from 12 to 150 horsepower at 400 pounds per square inch, depending on the size of the combination. The machine used in ordnance installations is designed to be run regularly at pressures of 1000 to 1500 psi, with intermittent service

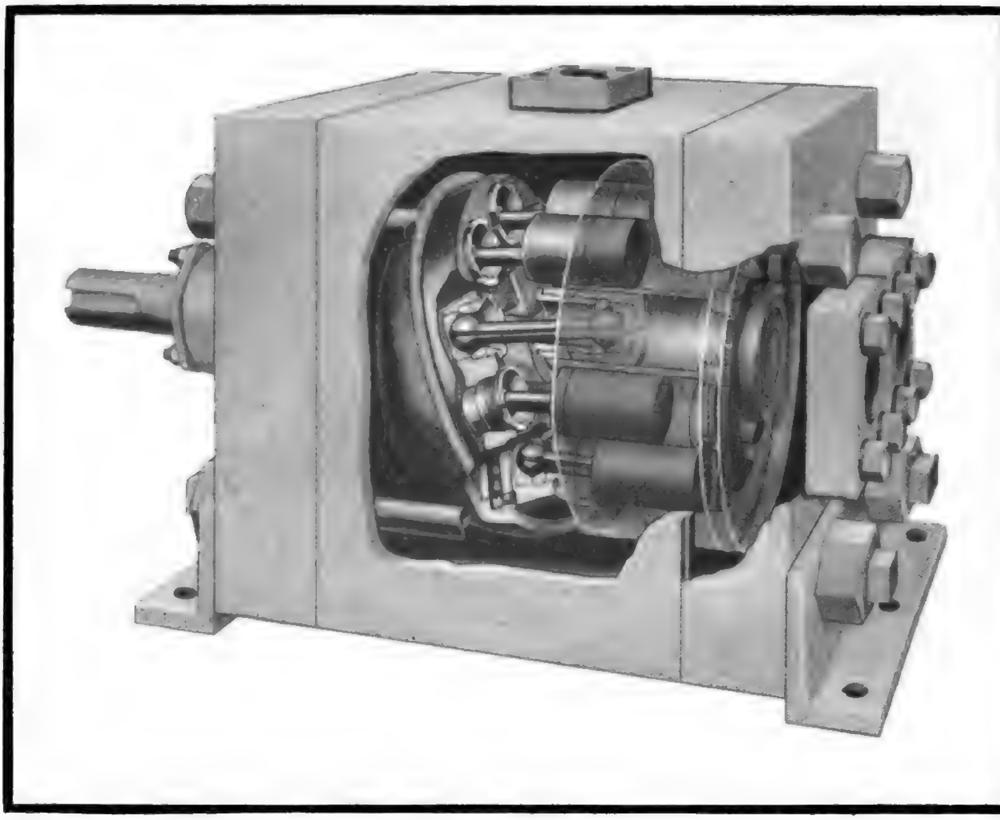

Figure 235

up to 2500 psi, while commercial designs are not supposed to be run regularly at pressures in excess of 500 pounds.

Northern Series 7000 Pump-Motor

The full description of the Waterbury pump and pump-motor combination that has been given renders unnecessary an equally detailed description of Northern axial piston pumps. They act according to the same principle and are similar even in details of construction. In some Northern axial piston pumps and pump-motors, however, the Williams universal joint is employed, and in others the Bendix-Weiss joint. Figures 234 and 235 show the pump and motor ends of the Northern combination.

Vickers Axial Piston Pump

Figure 236 illustrates this pump. It acts according to the same basic principle as the Waterbury and Northern pump, but the method of

accomplishing the action is entirely different, as will be clear from the following comparison.

First: Waterbury and Northern control the volume and direction of flow by changing the degree and direction of tilt of the A-end tilting box, whereas in Vickers this is done by pivoting the cylinder block to either side of a neutral position, as shown in Figure 237. In order to make this possible the cylinder block and valve plate are supported in a casting called a yoke. One end of the yoke is supported in the A-end case by pintles which furnish the required pivot point.

Second: In Waterbury and Northern the main transmission lines are connected to the valve plate, but in Vickers, since the valve plate moves from side to side with the cylinder block this would necessitate flexible transmission lines. It is therefore necessary to take the lines from the yoke pivot point. The yoke is cast with passages so

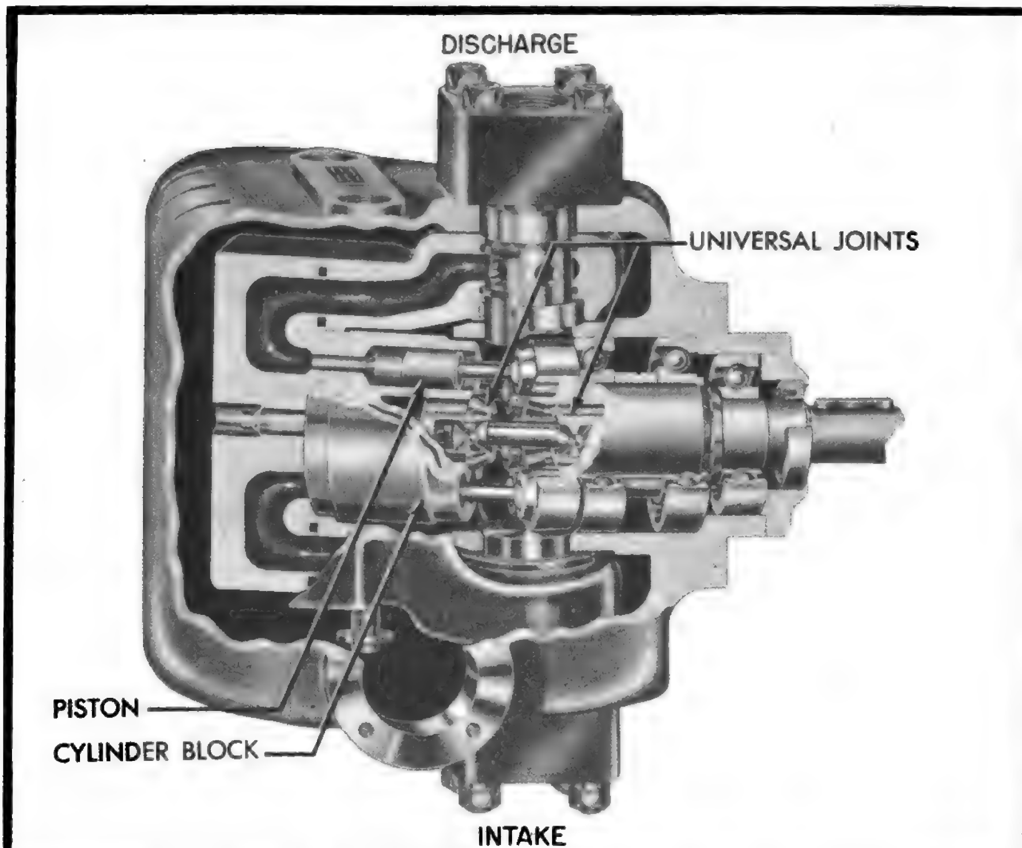

Figure 236

that oil set in motion in the cylinders and forced out through the valve plate is ported to the pintles, where it is passed out through the hollow pintle centers to the main transmission lines.

Third: Vickers does not have a socket ring. The connecting rods are connected to the shaft flange by ball and socket joints. In the small units the shaft flange and drive shaft are forged in one piece. In large units the shaft flange is force fitted and keyed to the end of the drive shaft.

Fourth: In all three units the power required to move the pistons is transmitted from the drive shaft to the pistons through the shaft flange (Vickers) or the socket ring (Waterbury and Northern) and the connecting rods. In the Waterbury and Northern the power is transmitted from the drive shaft to the socket ring through the universal joint. Therefore, the joint must be designed to carry this load. In the Vickers the universal link does not carry the load since it does not connect the drive shaft to the shaft flange. In this unit it connects the drive shaft to the cylinder block.

Uses of These Pumps

All three of the pumps covered in this chapter—the Waterbury, Northern 7000, and Vickers—are used in the Navy for the same

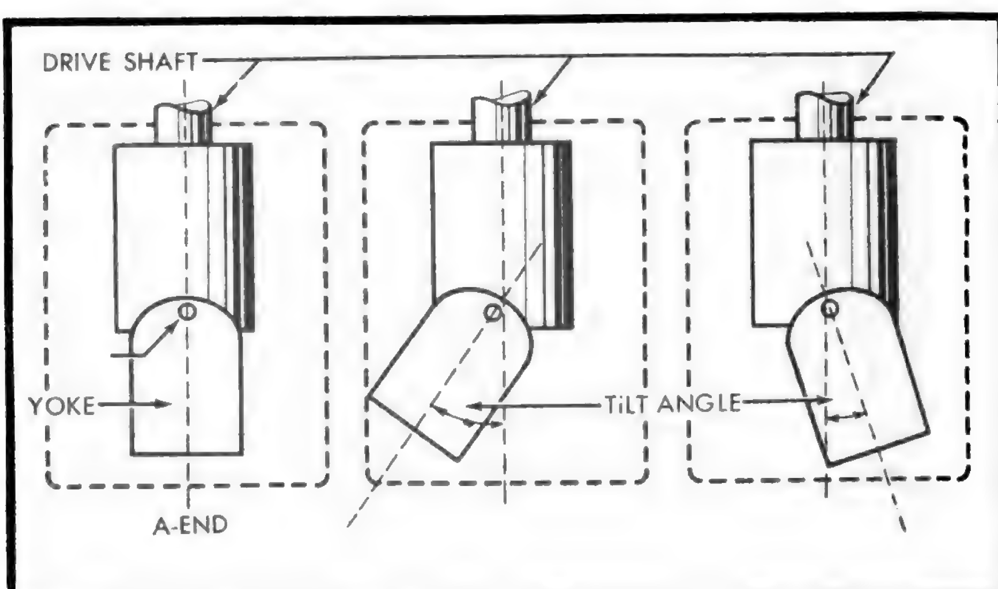

Figure 237

purposes to give precise control of steering operations, to actuate capstans, windlasses, etc., for gun and turret control, for flood and vent valves in submarines, and in other situations where exact control of movement is desired. Axial piston pumps are the only type approved for turret train and elevation units, on account of the great precision of operation there required. These uses will be further considered in Chapter 14.

Installation, Maintenance and Casualties

The installation and maintenance considerations presented in previous chapters apply to axial piston pumps. Even more than other pumps their successful operation depends on the care taken of them.

Cleanliness. Axial piston pumps require an extremely clean hydraulic system for completely satisfactory operation. The pipe and fittings should be carefully pickled on installation, or whenever their condition requires it. The position of the tubing or piping in the pickling bath should be changed occasionally to make sure that if gas pockets form, they will not always be at the same location. Precautions should be taken in pickling brass or copper pipe or tubing with steel fittings. Ordinarily the pipe or tubing will be pickled clean before the steel fittings are attached, but if heavy scale is encountered, it may be necessary to protect the steel fittings with paraffine (see Chapter 4).

Fluid to be used. The fluid used should conform to the manufacturer's specifications, and must be clean. It should not contain any light volatile oils to evaporate and leave air pockets. N.S. 2110 can be used with both Waterbury and Northern combinations (see Chapter 13). Where very low temperatures are expected, ice machine oil (N.S. 2075) may be used temporarily. For ordnance hydraulic units, O.S. 1113 and 2943 should be used, the latter where low temperatures are expected.

The fluid should completely fill the active parts of the mechanism, leaving no air pockets. The air valves are opened one or two turns during filling, so that air can escape through them to the

oil expansion box. Afterwards they are kept tightly closed. A supply of fluid must always be maintained in the oil expansion box.

A pounding or rattling noise due to partial vacuum produced in the active fluid may be unavoidable during high speed operation or under heavy loads. It can be ignored if it stops when the load is reduced. If not, air needs to be vented from the system.

Couplings. When the pump-motor combination is under load, there is absolutely no play back and forth in the shafts. For this reason a flexible form of coupling should be used to connect the B-end output shaft to the mechanism which it drives. Bevel gear connections are to be avoided only if one of the bevel gears is mounted on the B-end output shaft and would thereby produce a side thrust load on the shaft. If the bevel gears are housed in a separate box and the gear thrust loads are carried entirely by the box, and if a flexible coupling connects the B-end to the bevel gear shaft, the resulting combination would be completely acceptable.

Cooling. Where the pump-motor combination is to run continuously, some means of cooling the system is necessary, as by air or water circulation around the motor. If desired, the oil can be circulated through a cooling tank.

Air in the system. Noisy operation or jerky motion of the B-end may be due to air in the system. It should be vented.

Injured parts. If foreign matter has scored valve plates, pistons or cylinders, they may need to be reground or replaced. The cause of the scoring must also be removed. Pistons may seize if improperly fitted, and may have to be replaced or reground. The clearances are so small that pistons will work properly only if placed in the right cylinders. When the units are disassembled, the parts should be carefully marked so that they can be replaced properly.

QUESTIONS

1. For what purpose is the Williams universal joint used in an axial piston pump?
2. What mechanical irregularity is to be found in this joint?
3. How is this irregularity overcome in different kinds of axial piston pumps?
4. How does the Bendix-Weiss universal joint secure constant velocity?
5. What is the purpose in setting the tilting box of an axial piston pump at an angle?
6. What purpose do the lands on the valve plate of an axial piston pump perform?
7. Why are replenishing valves used with these pumps?
8. How are the air valves on these pumps operated?
9. How do axial piston motors differ from the pumps used with them?
10. Under what arrangement will the pump-motor unit not deliver power, although the pump is running?
11. How does the action of the Vickers axial piston pump differ from that of the Waterbury pumps?
12. For what purpose are these pumps used?
13. Why must particular care be taken with axial piston pumps to keep the system clean?

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Chapter 13

HYDRAULIC LIQUIDS

This chapter discusses: the functions performed by a hydraulic liquid; the properties it should possess; specifications—commercial and naval—for fluids used in hydraulic systems; the fluids used in hydraulic systems; the fluids recommended for naval use; and factors to be taken account of in the care and maintenance of hydraulic fluids.

Functions of a Hydraulic Liquid

Liquids are used in hydraulic systems primarily to transmit and distribute forces. As was pointed out in Chapter 1, they are able to do this because they are almost incompressible. As stated in Pascal's Law, a force applied on any area of an enclosed liquid will be transmitted equally and undiminished to all equal areas throughout the enclosure. Hence if a number of passages are open in a system, pressure can be distributed through all of them by means of the liquid.

In addition almost all liquids lubricate the parts they touch. This is especially the case with oils.

Liquids used. Water was used in the first hydraulic systems. It is still in use today in large commercial hydraulic installations suited to high pressures and low operating speeds, as for example for shipyard keel-benders or large freight elevators. Such systems may

require several thousand gallons of water for their operation, and may combine a low degree of mechanical efficiency with economy in terms of operating costs.

When water is used today in hydraulic systems, an emulsifying oil in a concentration of about two or three per cent by weight is usually added as a protection against rust and corrosion.

In modern naval ordnance hydraulic systems oil is almost invariably the hydraulic medium, since its lubricating qualities help to insure smooth, positive operation for fast moving parts. The selection of the correct hydraulic fluid is extremely important if a system is to work with maximum efficiency and have a long life.

Commercial manufacturers of hydraulic devices usually specify the type of fluid best suited for use with their equipment, in view of the working conditions, the service required, temperatures expected inside the system and out, pressures the fluid must withstand, the possibility of corrosion, etc. In naval ordnance systems, proper specifications are determined by the Bureau of Ordnance on the basis of experiments, tests and trials. The Bureau experienced many difficulties, for example, in the search for hydraulic fluids adapted to sub-zero Arctic climates and to the high temperatures of the tropics.

Properties of a Satisfactory Hydraulic Liquid

If incompressibility and fluidity were the only qualities desired, any liquid not too thick might be used in a hydraulic system. But a satisfactory liquid for a particular installation must possess a number of other properties. These we shall now discuss.

Chemical stability. All oils tend to undergo unfavorable chemical changes under severe operating conditions. This is the case, for example, when a system operates for a considerable period at high temperatures. A shearing action may take place when the oils must pass through small openings or between fast moving surfaces. They may "break down" if exposed to air, water, salt, or other

impurities, especially if they are in constant motion or subjected to heat. These chemical processes result in the formation of sludges, gums and carbon or other deposits, which clog openings, cause valves and pistons to stick or leak, and give poor lubrication to moving parts.

Freedom from acidity. An ideal oil would be free from acid and non-corrosive, so that it would not adversely affect the metals composing the system. But straight mineral oils cannot be expected to remain completely non-corrosive under severe operating conditions.

Certain chemicals, called *additives* or *inhibitors,* are sometimes dissolved in an oil to improve its chemical or physical behavior. Opinion is divided, however, on their merits, since they often work satisfactorily only for a limited period, after which the oil is subject to a more rapid deterioration than would otherwise have been the case. Yet when operating conditions are severe, as is often the case aboard ship, the use of additives may become almost unavoidable. The best procedure, however, is to use the fluid at hand, and then to protect the fluid and the system so far as possible from contamination by foreign matter, from abnormal temperature and from casualties and misuse.

Lubricating power. If motion takes place between surfaces in contact, friction tends to oppose the motion. When pressure forces the liquid of a hydraulic system between the moving surfaces, however, the liquid spreads out into a thin film which enables the parts to move more freely. The area is lubricated. Different liquids, including oils, vary greatly not only in their lubricating ability, but also in *film strength,* which is the capacity of a liquid to resist being wiped or squeezed out from between surfaces when spread out in an extremely thin layer. Film strength can be improved as is done in ordnance hydraulic fluids by the addition of certain additive materials. An oil will no longer lubricate if the film breaks down, since the motion of part against part wipes the metal clean of oil. Since lubricating power varies with temperature, climatic and working conditions must enter into a determination of the lubricating qualities of an oil.

Since this text is concerned only with liquids used for hydraulic purposes, it does not deal with oils and greases designed only for the lubrication of ordnance equipment. O.D. 3000 discusses the proper lubricants to be used with nearly all ordnance equipment, including the substitutes that may be used when the designated lubricant is not available. It should be consulted frequently. As lubrication materials and procedures are improved, the Bureau of Ordnance publishes circular letters, which should be filed for reference as occasion demands.

Viscosity. Some liquids are thinner than others and therefore flow more easily. The thinner liquids are called less viscous. In the strict sense viscosity is defined as *the resistance which a liquid offers to flow.* Viscosity increases with decreases in temperature, since liquids always flow less easily when cold than when hot.

A satisfactory fluid for a given hydraulic system must have enough body to give a good seal at pumps, valves and pistons, but must not be so thick that it offers excessive resistance to flow, leading to power loss and higher operating temperatures. These factors will add to load and to excessive wear of parts. On the other hand, however, too thin a fluid will also lead to unnecessarily rapid wear of moving parts, or of parts which may have heavy loads.

Measurement of viscosity. Although it hardly conforms to scientific standards of accuracy, the instrument most often used by American engineers to measure the viscosity of liquids is the Saybolt Universal Viscometer or Viscosimeter (Figure 238). This instrument measures the number of seconds it takes for a fixed quantity of liquid (60 cubic centimeters) to flow through a small orifice of standard length and diameter at a specified temperature. The viscosity is stated as so many units S.S.U., or Seconds, Saybolt Universal, at such and such a temperature. For example, a certain oil might have a viscosity of 80 S.S.U. at 130° *F*.

The Saybolt Universal Viscometer consists of a reservoir for the oil surrounded by a bath heated by heating coils to bring the oil to the temperature at which the viscosity is to be measured. The

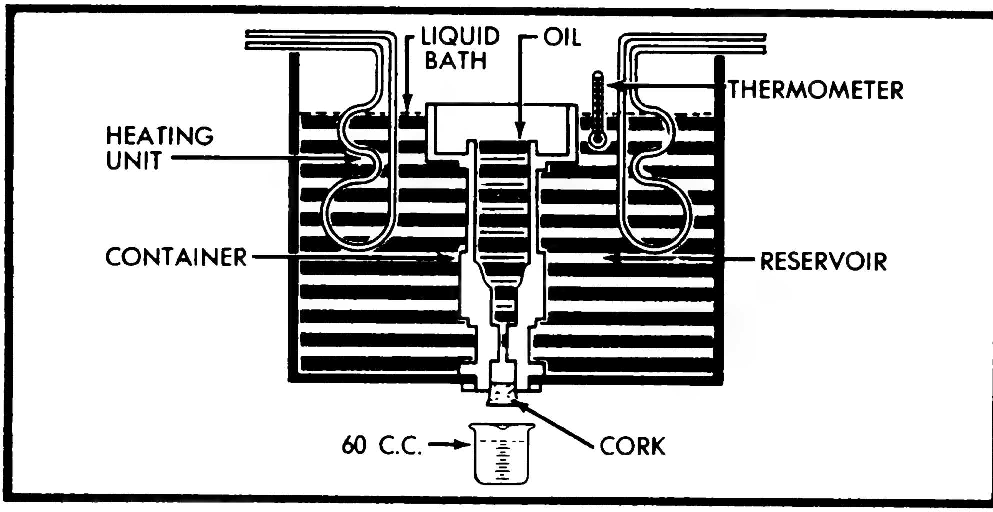

Figure 238

bottom of the reservoir issues into the standard viscometer orifice. Passage through the orifice is blocked by a cork. The reservoir is filled to a marked level with oil and a container marked at the 60 cc level is placed under the opening. When the oil to be tested is at the desired test temperature, the cork is removed, and the number of seconds it takes for the liquid to reach the 60 cc level gives the S.S.U. reading.

Navy Symbol Oils. The Bureau of Ships identifies each oil used on shipboard by a four-digit symbol, the first digit indicating the class of oil and the last three digits giving its approximate viscosity at a stated temperature—210° *F* for oils of classes 1, 3, 4, 5, 6, and 7, 130° *F* for those of classes 2, 8, and 9.

Thus N.S. 1042 is an oil of class 1 having a viscosity of 40–44 S.S.U. at 210° *F*, while N.S. 2110 is an oil of class 2 with a viscosity of 90–120 S.S.U. at 130° *F*. An oil of class 1 is described as being a 1000 series Navy Symbol Oil; of class 2 as 2000 series; etc.

Commercial classification. The Society of Automotive Engineers also classifies oils according to viscosity, so that consumers may select the proper lubricant for a particular use without respect to brand name or place of origin, but its classification is a very

broad one. Thus, S.A.E. 20 has a viscosity of 120–185 S.S.U. at 130° *F*, and S.A.E. 30 has a viscosity of 185–255 S.S.U. at the same temperature.

Viscosity index. The viscosity of an oil decreases as the temperature increases, or in other words the oil becomes thinner, but the variation is greater for some oils than for others. Thus Pennsylvania crude oils (paraffinic) vary comparatively little in viscosity with changes in temperature, while with Gulf Coast crudes (naphthenic) the variation is considerably greater.

In order to obtain a numerical indication of the degree to which viscosity changes with changes in temperature, these two oils are taken as the basis for a scale. The *change* in viscosity of Pennsylvania crudes between 100° *F* and 210° *F* is rated as *100;* while the change in viscosity of Gulf Coast crudes over the same temperature range is rated as *0.* Other oils are then assigned a viscosity index in terms of the degree to which their viscosity changes over this range, as compared with the standard oils.

The lower the V.I. (viscosity index), the greater the variation in viscosity with changes in temperature. V.I. figures may range above 100 or below 0, if the oils being measured vary less or more widely in viscosity than the standard oils. Thus an oil with a viscosity index of —10 would vary in viscosity with changes in temperature to an even greater degree than Gulf Coast crudes; while an oil with a viscosity index of 120 would show even less change in viscosity with changes in temperature than Pennsylvania crudes.

Since naval hydraulic systems must work satisfactorily under wide temperature extremes, from Arctic regions to the Tropics, the liquids used should have as high a viscosity index as possible, consistent with the other properties the fluid must possess. The viscosity index of an oil may be increased through the use of chemical additives.

Pour point. This is the temperature at which an oil will congeal or solidify in a standard container. It varies from oil to oil accord-

ing to the nature of the crude oil used, the refining methods used, and the viscosity of the product. Chemical additives can be used to lower the pour point of oils. The addition of a pour-point depressant will probably affect viscosity insignificantly through the entire temperature range. The pour point is lowered, but the viscosity continues to increase with decrease in temperature at the same rate it did before the pour point was reached.

Any fluid used for hydraulic purposes should have a pour point well below its minimum operating temperature.

Flash point. This is the lowest temperature at which a liquid gives off vapor in sufficient quantity to ignite momentarily or "flash" when a flame is applied. The flash point of the fluid is desired to be high because it indicates that evaporation will be low, i.e.: evaporation at normal temperatures. With O.S. 2943 with a flash of 225° *F*, the operating temperature of the equipment will exceed the flash point at various points of the system. There is no evaporation because there is pressure at these points.

Oils Recommended for Naval Use

Oils for relatively protected systems. It is probably impossible to find a fluid perfectly suited in all respects for use in naval ordnance hydraulic systems. Bureau of Ships' steering gear and control apparatus are located where they are protected from extremes of temperature and weather as a rule. N.S. 2110 or N.S. 2135 were specified by the Bureau of Ships for use under these conditions. The characteristics of these oils are as follows:

	Viscosity at 130° F	*Comparable to*	*Viscosity index*	*Flash point*	*Pour point*
N.S. 2110	90–120 S.S.U.	S.A.E. 10	Relatively low	325° *F*	0° *F*
N.S. 2135	120–145 S.S.U.	S.A.E. 20	Relatively low	340° *F*	0° *F*

N.S. 2135 is not suitable for exposed locations nor for a wide range of operating conditions, since it may have a low viscosity index.

Fluids for open mounts: O.S. 1113. As hydraulic systems were

applied to open mounts, it became necessary to develop a fluid suitable for use under wider extremes of temperature—one, that is, with a high viscosity index. O.S. 1113 is such a fluid—"O.S." standing for *Ordnance Specification,* since this fluid is not listed in the N.S. (Navy Symbol) classification of the Bureau of Ships.

O.S. 1113 has about the viscosity of N.S. 2110 at about 70° *F*, but is a little thinner at low temperatures and a little thicker at high. It has a pour point of minus 40° *F*, and a minimum flash point of 300° *F*, which makes it safe to use in all ordnance hydraulic systems. These highly desirable properties are attained through the use of chemical additives which do not affect the desirable properties of the fluid. The fluid should however be checked periodically, for visible presence of water or dirt and for neutralization number to determine if renovation is necessary or feasible. The renovated fluid should meet the specifications of the original fluid. The fluid should therefore be tested periodically for sludge or carbon, and renovated or replaced when necessary. Testing facilities are available at all Navy Yards.

O.S.2943. When naval ordnance hydraulic systems had to be started in extremely cold weather, it was found necessary to keep the temperature of O.S. 1113 above 35° to 45° *F* to insure operation. O.S. 2943 was therefore developed. It is much less viscous than O.S. 1113 at low temperatures, and of approximately the same viscosity at 160° *F*, after which it is progressively more viscous than O.S. 1113.

It has a considerably greater viscosity index than O.S. 1113. Therefore, it varies less in viscosity than that fluid. The following table gives some comparisons of actual fluids:

	Viscosity at 0 F S.S.U.	*Viscosity at 100 F S.S.U.*	*Viscosity at 210 F S.S.U.*
O.S. 1113	5100	195	55
O.S. 2943	990	132	60

O.S. 2943 also has superior rust inhibiting qualities and is able to take up small amounts of water introduced into the system by con-

densation. Tests have demonstrated satisfactory operation at minus 10° *F*, as compared to plus 25° *F* for O.S. 1113. It also operates satisfactorily for long periods at 180° *F*, whereas difficulties may arise with O.S. 1113 when the fluid temperature exceeds 170° *F*.

If temperatures below minus 10° *F* are encountered, the equipment should be exercised under power for 15 minutes or more per hour, as needed, to keep the temperature high enough to permit free operation.

Substitutes. O.S. 2943 is now the accepted liquid for all ordnance hydraulic systems. If it is not available, O.S. 1113 is the best substitute. If this too is not at hand, the following fluids may be used under the conditions specified, but the substitute used should be replaced by O.S. 2943 as soon as it is available:

N.S. 2110.—Is usable under summer conditions in cases of emergency only. The characteristics of this fluid have already been given above. Equipment will usually require exercising to achieve satisfactory operation below 40° *F*.

Specification 14–0–12 (transformer oil) formerly N.S. 9045—may be used as an emergency substitute when low temperatures require a lighter oil than N.S. 2110.

O.S. 2943 is a hydraulic fluid, and not a lubricating oil. It lubricates satisfactorily when used for operating purposes in closed hydraulic systems, but should not be used to lubricate in thin films or on exposed parts because, due to evaporation this fluid has a tendency to dry to a sticky residue when exposed to the air.

Care and Maintenance of Hydraulic Fluids

Cleanliness. The principal factor in the maintenance of a hydraulic system is cleanliness. The most usual and avoidable cause of trouble is the presence of foreign matter that has entered the system through carelessness of personnel. The large number of small orifices and close clearances makes all forms of dirt a positive hazard. It pays in time saved, as well as in efficient operation, to keep a system clean.

To illustrate the importance of clean liquid in hydraulic mechanisms, airplanes are equipped with filters which exclude particles having dimensions greater than 1/5000 inch. Such particles would take two days to settle one foot in fresh water.

Fluid being added to a system should be filtered through the Luber-Finer Hydraulic Fluid Filter, Model 750–NS–1. This equipment is a completely self-contained portable oil filter. The equipment is provided with electric cables which may be plugged into a 115-volt AC supply, and inlet and outlet hoses are supplied of convenient length to reach the equipment or containers which hold the fluid to be filtered. When the filter element is new, the hose relatively short, and the temperature of the fluid and equipment is in the range of 70° to 90° *F*, 3 gallons of hydraulic fluid per minute can be filtered with an indicated pressure of 25 to 30 psi.

When the Luber-Finer filter is not available, fluid should be filtered through a screen of at least 200 wire mesh. It is available in sheets at all Navy Yards. A circular piece of the screen can be soldered near the bottom of a large funnel (Figure 239), so that the head of liquid over the screen will help to force the liquid through. This funnel should be used only for straining hydraulic fluid, and should be fitted with a cover and a cork to exclude foreign matter when it is not in use.

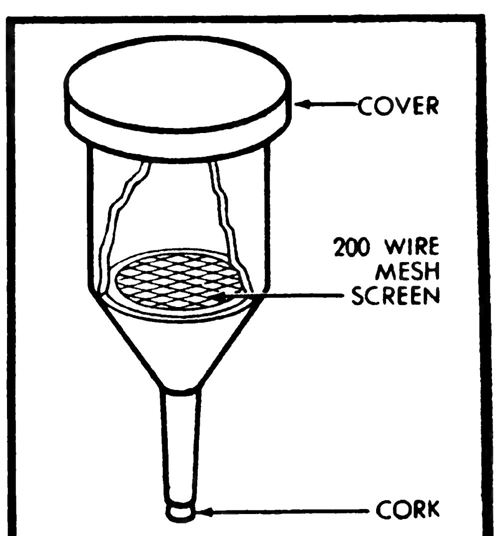

Figure 239

Flushing hydraulic units. When foreign matter or sludge is found in a system, it is usually possible to clean the units without disassembly. Diesel fuel oil of the proper specifications is the best available flushing compound. Diesel fuel oil, Bureau of Ships Specification 7–0–2 (INT) (Grade 2), is recommended for flushing purposes. The oil is circulated in the system under low power and low pressure in order

to clean sludge and dirty fluid from old systems or heavy oils and rust preventive compounds from new systems.

The following procedure for cleaning the system is recommended:

1. Drain out the old fluid as completely as possible.
2. Close all connections and fill the system with diesel fuel oil.
3. Operate the unit smoothly under local control, and through very small increments in order to fill the system thoroughly, and allow the system to stand for about an hour.
4. Operate the equipment under power for 3 to 5 minutes under low load and smooth operation in local control, and allow it to stand idle for 15 minutes. Repeat this process two or more times. If the system contains "Cuno" filters, the cleaning handle of these units should be turned, and an amount of oil about equal to the volume of the filter drained off after each operating period.
5. If time is available, allow the system to stand idle for about an hour. The diesel fuel oil should then be drained as completely as possible, and the system refilled with the proper hydraulic fluid.
6. After operating the system for one or two days with this charge of hydraulic fluid, it is preferable, but not essential, to drain the system and refill with a new charge of hydraulic fluid. If only one system is being cleaned, the procedure of changing fluid after two days is impractical on account of the quantity of fluid that would be wasted. If a number of systems are to be cleaned, however, the same initial charge of hydraulic fluid can be used in successive units to flush out the diesel fuel oil.

In cleaning a system, the following precautions are to be observed:

1. The hydraulic units must not be put under full load while being flushed with diesel fuel oil.
2. The operating pressure must be kept as low as possible.
3. The unit must not be run at any one time for a longer period than 5 minutes with diesel fuel oil in the system.

Disassembly. When a system is fouled from corrosion and foreign matter not soluble in diesel fuel oil, it will be necessary to disassemble the unit and thoroughly clean, repair or replace parts, as necessary. If units of a system must be disassembled on weather

decks, a canopy should be rigged to protect them from moisture and foreign matter.

Storing hydraulic units. When a unit is not to be used for a relatively long period, it can be filled with a chemical preservative and then drained, so that a thin protective film is formed on all surfaces to protect them against rust and corrosion. Thin Film Rust Preventive Compound (Polar Type), Bureau of Ships Specification 52–C–18 (INT), is recommended for this purpose. This specification (52–C–18) provides for three grades of the compound and each grade is designed for a particular use. Grade 1 is intended for equipment to be stored outside and forms a thicker, tougher and more weather resistant film than Grades 2 or 3. Grade 2 is intended for corrodible metal surfaces where under-cover storage is provided. Grade 3 is intended for use on corrodible metal surfaces to displace residual moisture, form an emulsion with the water and lay down a temporary protective film. An important use for Grade 3 is found in salvage work to prevent further rusting of equipment that has been submerged in water, pending disassembly for overhaul.

The Bureau of Ordnance recommends Grade 2 compound for inside, inactive storage of most types of hydraulic equipment. The system is filled with the liquid, operated for a few minutes and then drained. When the equipment is being prepared for active service the hydraulic system must be thoroughly flushed with diesel fuel oil, Federal Specification VV–L–791, 1 June 1940, Bureau of Ships Specification 7–0–2 (INT). Run the equipment with the system filled with diesel fuel oil and observe the following precautions:

1. The hydraulic unit should not be placed under full load while being flushed with diesel fuel oil.
2. The operating pressure required should be kept as low as possible.
3. The unit should not be run at any one time for a period greater than 5 minutes with diesel fuel oil in the system.
4. Thoroughly drain diesel fuel oil out before filling the system with hydraulic fluid.

While Grade 2 has been used with perfect safety for most hydraulic

systems, it is not recommended for use in any system with a Vickers Vane Pump (Figure 198). The Bureau of Ordnance recommends that these systems be preserved with rust inhibiting oil O.S. 1363.

QUESTIONS

1. What is lubrication?
2. Why does sludge form in an oil?
3. What is viscosity, and how can it be expressed?
4. How do temperature changes affect the viscosity of an oil?
5. Explain the following symbols: N.S.; O.S.; S.A.E.
6. What is the viscosity index of an oil? The pour point? The flash point?
7. What fluids are now specified by the Bureau of Ordnance for use in naval hydraulic systems? What fluids may be used as substitutes?
8. What is the most important single factor in the care of any hydraulic system?
9. How is a hydraulic system flushed?
10. What is the accepted procedure for preparing hydraulic units for storage?
11. What information is to be found in O.D. 3000?

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Chapter 14

ORDNANCE HYDRAULIC SYSTEMS

In this chapter the large variety of systems used in ordnance hydraulics are grouped according to basic points of similarity, and their fundamental modes of operation are illustrated and briefly described.

The many systems used in ordnance hydraulics can be classified as follows:

1. Recoil and counterrecoil systems, which require only a piston and cylinder and hydraulic fluid (liquid and/or air), without a pump or pump-motor combination.
2. Constant delivery systems, which make use of a constant delivery positive displacement pump, driving either (a) a piston and cylinder or (b) a hydraulic motor.
3. Variable delivery systems, which make use of a variable delivery positive displacement pump, driving either (a) a piston and cylinder or (b) a hydraulic motor. Systems of the latter kind can be controlled: (i) manually; (ii) automatically by hydraulic means; (iii) automatically by electrical means.

Each of the systems listed above will be illustrated and described in order.

Recoil and Counterrecoil Systems

Recoil systems are designed to dissipate part of the energy of recoil when the gun is fired. They consist essentially of a piston in a

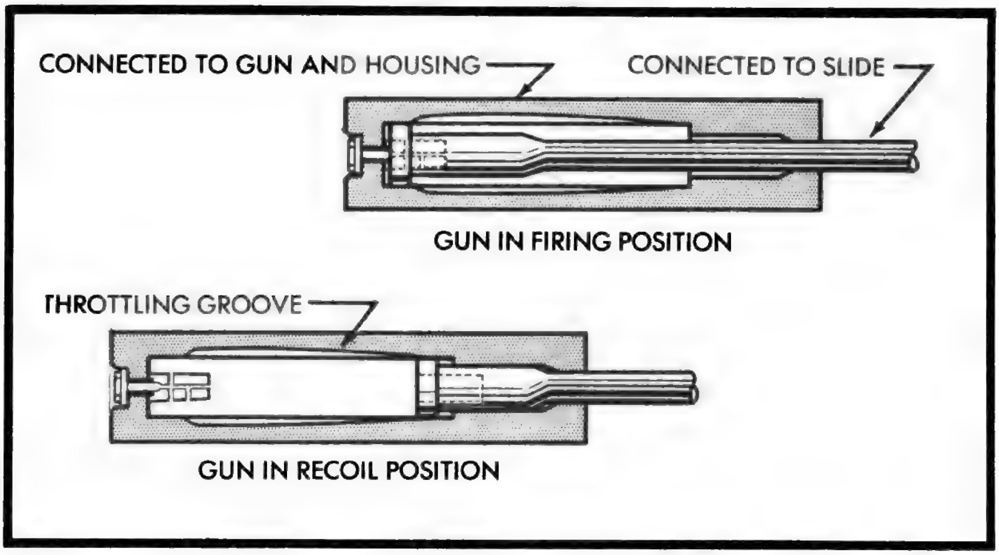

Figure 240

cylinder containing a hydraulic liquid (Fig 240). Either the piston or the cylinder is mechanically joined to the gun housing which recoils with the gun. The other element is mechanically joined to the slide on which the housing travels during recoil. In Figure 240, the cylinder moves back with the gun, while the piston stays in fixed position with the slide.

When the gun is fired, the recoil produces a relative motion of the barrel and gun housing with respect to the slide, and therefore of the piston with respect to the cylinder. This forces the liquid from one side of the piston to the other through specially shaped small passages, of which the throttling grooves in Figure 240 are an example. As a result a large part of the energy of recoil is dissipated in friction, and the rearward motion of the gun and housing is quickly brought to a stop. The rest of the energy of recoil is used to charge the counterrecoil system, where it is in turn dissipated in friction when the gun is returned to the firing position.

The distance of recoil varies from gun to gun, but is never great. In one 16-inch gun, for example, it amounts to 47 inches at full elevation, while the normal recoil of one 5-inch gun is 15 inches. Counterrecoil systems are designed to return the gun to battery position after the force of recoil has been dissipated. They con-

sist essentially of a plunger in a cylinder containing air under pressure (Figure 241). One of the two elements is mechanically joined to the gun housing, while the other is mechanically joined to the slide on which the housing travels during recoil. In Figure 241 the plunger is connected to the slide, while the cylinder is connected to the gun housing.

When the gun is fired, the relative motion of the plunger in the cylinder during recoil accumulates energy by further compressing the air. The expansion of the air after recoil has been completed forces the plunger back to its original position, and therefore returns the gun to battery position.

In some mounts loss of air from the counterrecoil system is prevented through use of a differential piston unit consisting of a piston and cylinder and a supply of hydraulic liquid, as shown

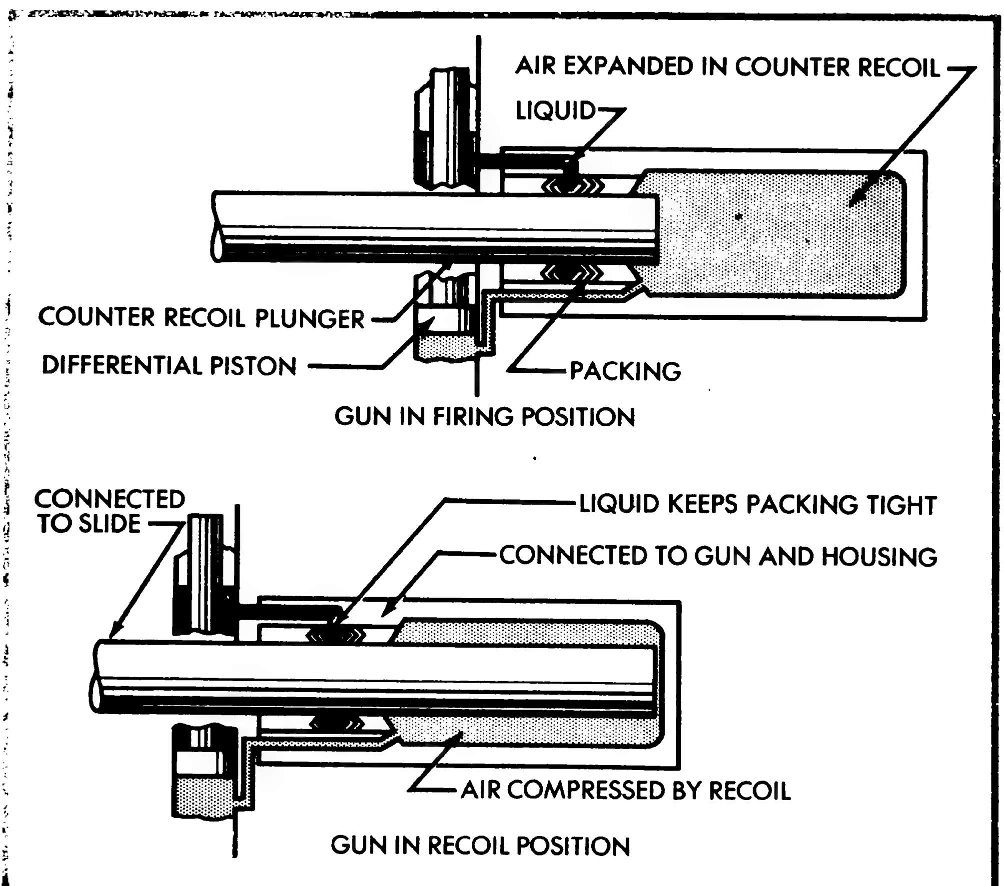

Figure 241

in Figure 241. The liquid is carried to one side of the differential piston, while the compressed air from the counterrecoil cylinder is carried to the other side of the piston. The liquid is also carried to a space in the packing of the counterrecoil plunger.

When the air in the counterrecoil cylinder is further compressed during recoil, the air in the differential cylinder is equally compressed, so that pressure against the air side of the differential piston is increased. (It should be pointed out that a gas under considerable pressure behaves hydraulically in the same manner as a liquid.) A total force equal to the air pressure times the area of the piston head is therefore applied to the liquid on the other side of the piston. But the area of the piston on the liquid side of the piston is less than on the air side, since the piston rod takes off from the piston on the liquid side. The force of the air pressure acting on the liquid therefore produces a greater pressure or force per unit of area there. Pressure in the liquid will always be considerably greater than the air pressure in the system. This is an application of the principle of differential areas explained in Chapter 1.

This liquid pressure is communicated to the space in the packing of the counterrecoil plunger. It acts there to hold the air in the cylinder. Since the liquid pressure is always greater than the air pressure, liquid may leak somewhat into the cylinder, but the compressed air will be held there.

It will be noted that recoil and counterrecoil systems make no use either of a pump or a pump-motor combination. All other ordnance hydraulic systems require one or the other. Recoil and counterrecoil systems also make little use of the valves. A needle valve is sometimes required to permit precise adjustment of the last portion of the counterrecoil action. These systems are also relatively less complicated mechanically than other ordnance hydraulic systems. At the same time their exact design involves very precise calculations. An obvious example is the exact determination of the number, size and precise shape of the passages through which the liquid flows from one side of the recoil piston to the other during recoil.

Constant Delivery Systems

As used in ordnance hydraulics, these systems consist of a constant delivery positive displacement pump, called the A-end, driven by a constant speed electric motor. The pump discharges a constant volume of fluid. The flow of the fluid is controlled by valves. Directional and check valves control the direction of flow, pressure regulating valves control the pressure in secondary lines (if there are any), and relief valves protect the system from excess pressure.

By means of the control mechanism the discharge of the pump is directed to the working unit, which is called the B-end. The working unit can be of two kinds. It can consist of a piston and cylinder, if the output requires a reciprocating motion; or it can consist of a hydraulic motor, if the output requires a rotary motion.

(a) *Constant delivery systems with piston and cylinder.* Figure 242 is a simplified block diagram showing the main elements of this kind of system in advance and retract positions. The manner of action should be entirely clear, since it has been fully explained earlier in this book.

An electric motor drives a pump so that fluid is forced from a reservoir into the discharge pipe of the pump. This fluid is carried to a control mechanism (for example a directional valve), which is positioned to deliver the fluid to one or the other side of a work piston. At the same time the control mechanism is open to receive fluid from the other side of the work piston and direct it to the reservoir. The fluid coming into the work cylinder is under high pressure, since it is acting against a resistance, while the other fluid in the work cylinder is under low pressure. The difference in pressure on the two sides of the work piston drives it towards the low-pressure end of the cylinder, discharging the fluid which is there through the control mechanism and on to the reservoir. In this manner one phase of a work operation is performed through mechanical linkages at the outside end of the piston rod.

A reversed positioning of the control mechanism will reverse the

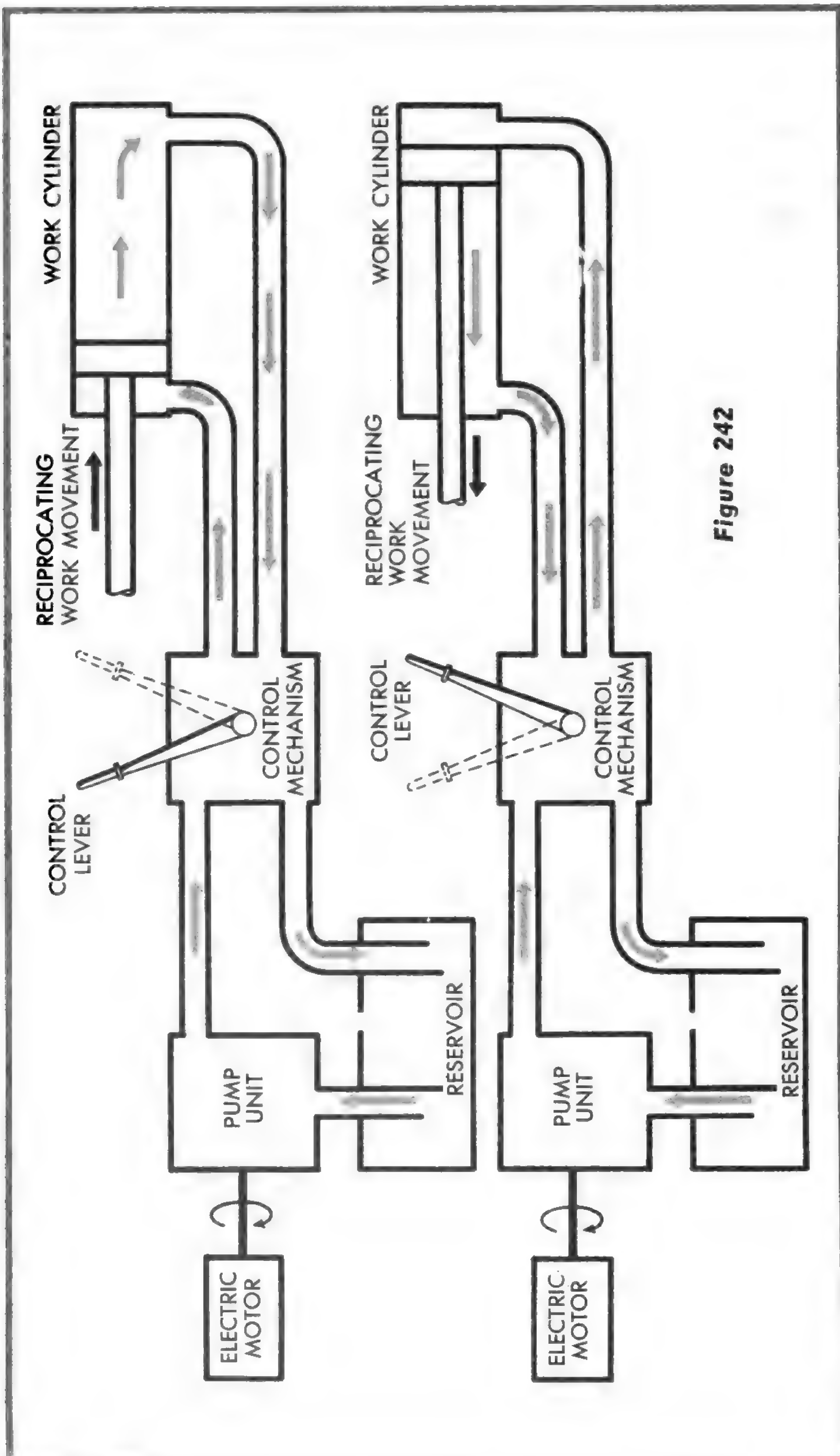
WORK CYLINDER
RECIPROCATING WORK MOVEMENT
CONTROL LEVER
CONTROL MECHANISM
PUMP UNIT
RESERVOIR
ELECTRIC MOTOR
WORK CYLINDER
RECIPROCATING WORK MOVEMENT
CONTROL LEVER
CONTROL MECHANISM
PUMP UNIT
RESERVOIR
ELECTRIC MOTOR

Figure 242

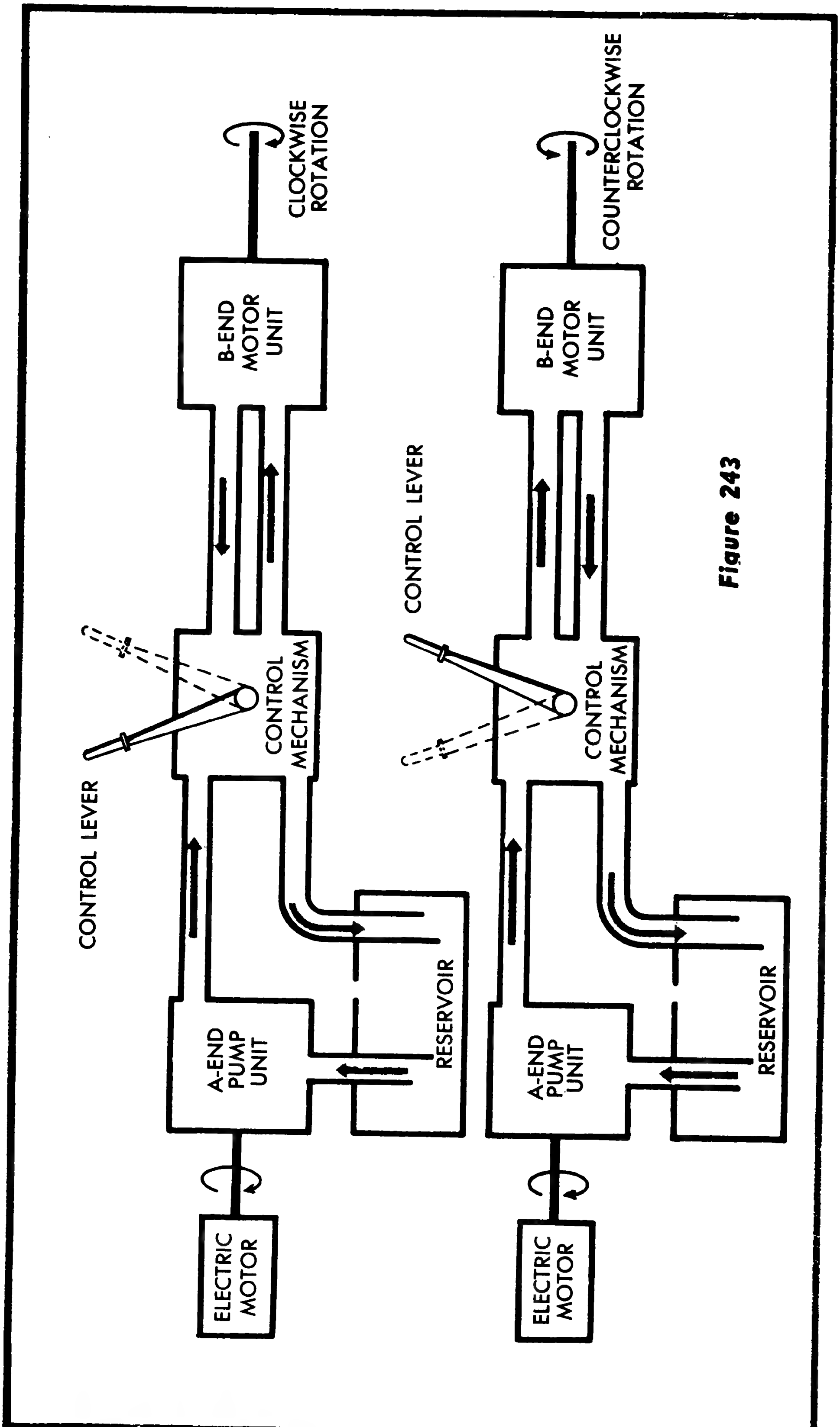
CLOCKWISE ROTATION
B-END MOTOR UNIT
CONTROL LEVER
CONTROL MECHANISM
A-END PUMP UNIT
RESERVOIR
ELECTRIC MOTOR
COUNTERCLOCKWISE ROTATION
B-END MOTOR UNIT
CONTROL LEVER
CONTROL MECHANISM
A-END PUMP UNIT
RESERVOIR
ELECTRIC MOTOR

Figure 243

flow of fluid to and from the work cylinder, so that the work piston will be moved in the opposite direction. In this manner the reverse phase of a work movement can be performed.

Constant delivery systems with piston and cylinder B-ends are used with rammers, breech mechanisms, cradle operating systems (for 12- and 16-inch guns), and power car door operating systems (for 12- and 16-inch guns).

(b) *Constant delivery systems with hydraulic motor.* Figure 243 is a simplified block diagram showing the main elements of this kind of system for clockwise and counterclockwise rotation of the work shaft. Once more the manner of action should be entirely clear.

An electric motor drives a pump so that fluid is forced from a reservoir into the discharge pipe of the pump. This fluid is carried to a control mechansm which is positioned to deliver the fluid to one or the other port of a hydraulic motor. At the same time the control mechanism is open to receive fluid from the other port of the hydraulic motor and deliver it to the reservoir. The fluid coming into the one port of the hydraulic motor is under high pressure, while the other fluid in the motor is under low pressure. The difference in pressure in the two areas in the hydraulic motor forces its pistons to reciprocate in their cylinders, and establishes the two ports as respectively high-pressure and low-pressure ports. The reciprocating motion of the pistons causes the cylinder block of the motor to rotate, and this in turn rotates the drive shaft to which the cylinder block is attached. In this manner one phase of a work operation is performed through mechanical gears and linkages connected to the drive shaft.

A reversed positioning of the control mechanism will reverse the flow of fluid to and from the hydraulic motor. Fluid will be delivered to what was previously the low-pressure port, and will be returned from what was previously the high-pressure port. The direction of rotation of the drive shaft of the hydraulic motor will be reversed, and in this manner the reverse phase of a work operation will be performed.

Constant delivery systems with hydraulic motor B-ends are used with rammers, powder and projectile hoists, and dredger hoists.

Variable Delivery Systems

As used in ordnance hydraulics, these systems consist of a variable delivery positive displacement pump driven by a constant speed electric motor. The pump is capable of discharging a variable volume of fluid at constant or varying pressures. Flow can be varied from a maximum volume of flow in one direction through zero flow, to a maximum volume of flow in the opposite direction.

The volume and direction of flow are controlled by a control mechanism which regulates the angle of tilt of the pump tilting box. Increases in the angle of tilt away from the neutral position increases the length of stroke of the pistons of the pump, and thereby increases the volume of flow. With the tilting box off neutral in one direction flow is produced in one direction, while departures from neutral in the opposite direction reverse the direction of flow.

The pressure developed in variable delivery systems depends on the work resistance and not on the tilt angle or volume output of the pump. Up to the capacity of the relief valves, the pressure will rise to meet any resistance. With a small tilt of the tilting box, we can secure the effect of a small volume high-pressure pump; with a large tilt, we can secure the effect of a large volume low-pressure pump.

By control of the tilting box, the discharge of the pump is directed to the B-end working unit. The working unit can be of two kinds. It can consist of a piston and cylinder, if the output requires a reciprocating motion; or it can consist of a hydraulic motor, if the output requires a rotary motion.

Constant and variable delivery systems can be combined to perform a single work operation. Or two constant delivery systems, one of high volume and the other of low, can be combined in operation to produce some of the flexibility of a variable volume system (Figure 244). Delivery of liquid from both pumps goes to the

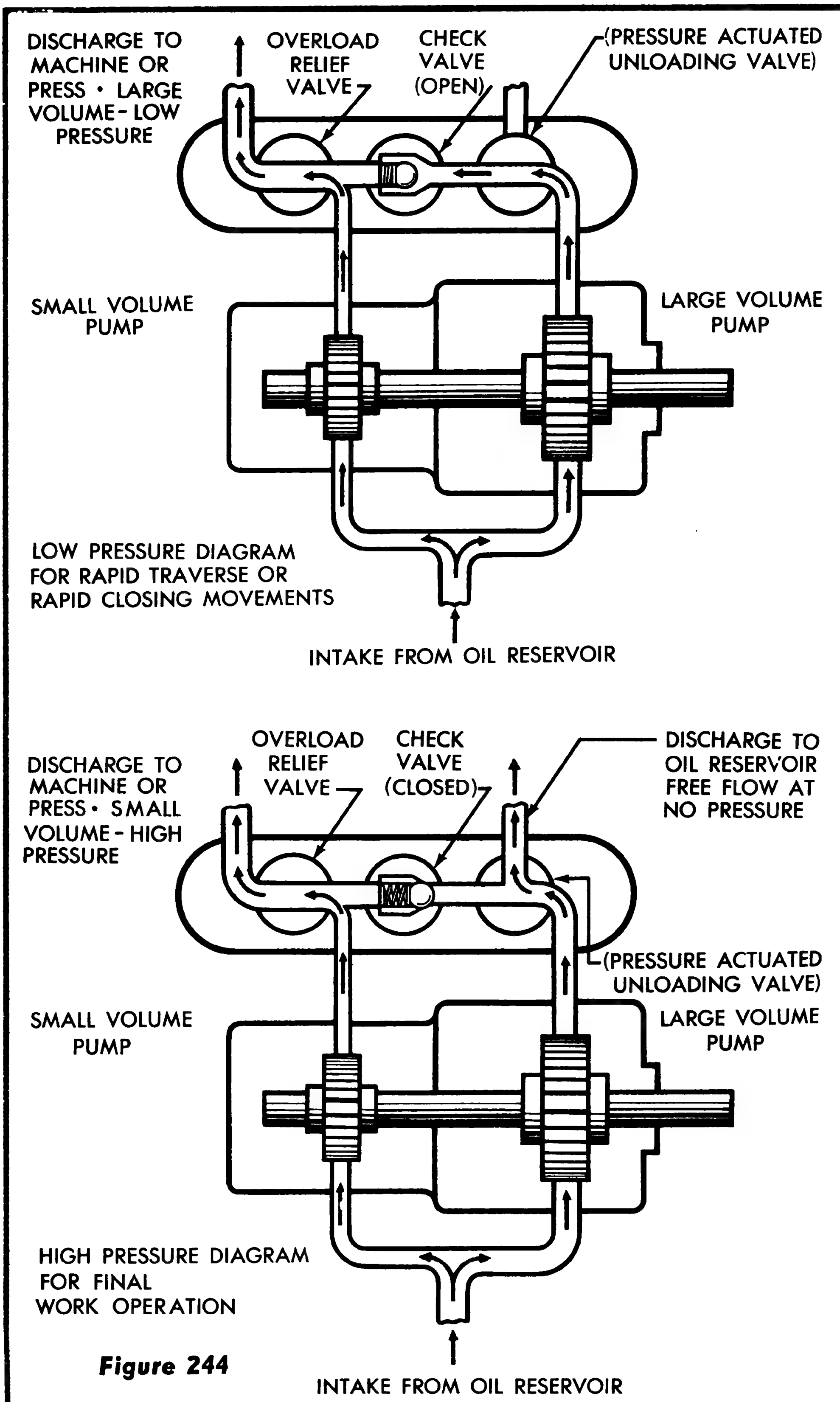
DISCHARGE TO MACHINE OR PRESS • LARGE VOLUME - LOW PRESSURE
OVERLOAD RELIEF VALVE
CHECK VALVE (OPEN)
(PRESSURE ACTUATED UNLOADING VALVE)
SMALL VOLUME PUMP
LARGE VOLUME PUMP
LOW PRESSURE DIAGRAM FOR RAPID TRAVERSE OR RAPID CLOSING MOVEMENTS
INTAKE FROM OIL RESERVOIR
DISCHARGE TO MACHINE OR PRESS • SMALL VOLUME - HIGH PRESSURE
OVERLOAD RELIEF VALVE
CHECK VALVE (CLOSED)
DISCHARGE TO OIL RESERVOIR FREE FLOW AT NO PRESSURE
(PRESSURE ACTUATED UNLOADING VALVE)
SMALL VOLUME PUMP
LARGE VOLUME PUMP
HIGH PRESSURE DIAGRAM FOR FINAL WORK OPERATION
INTAKE FROM OIL RESERVOIR

Figure 244

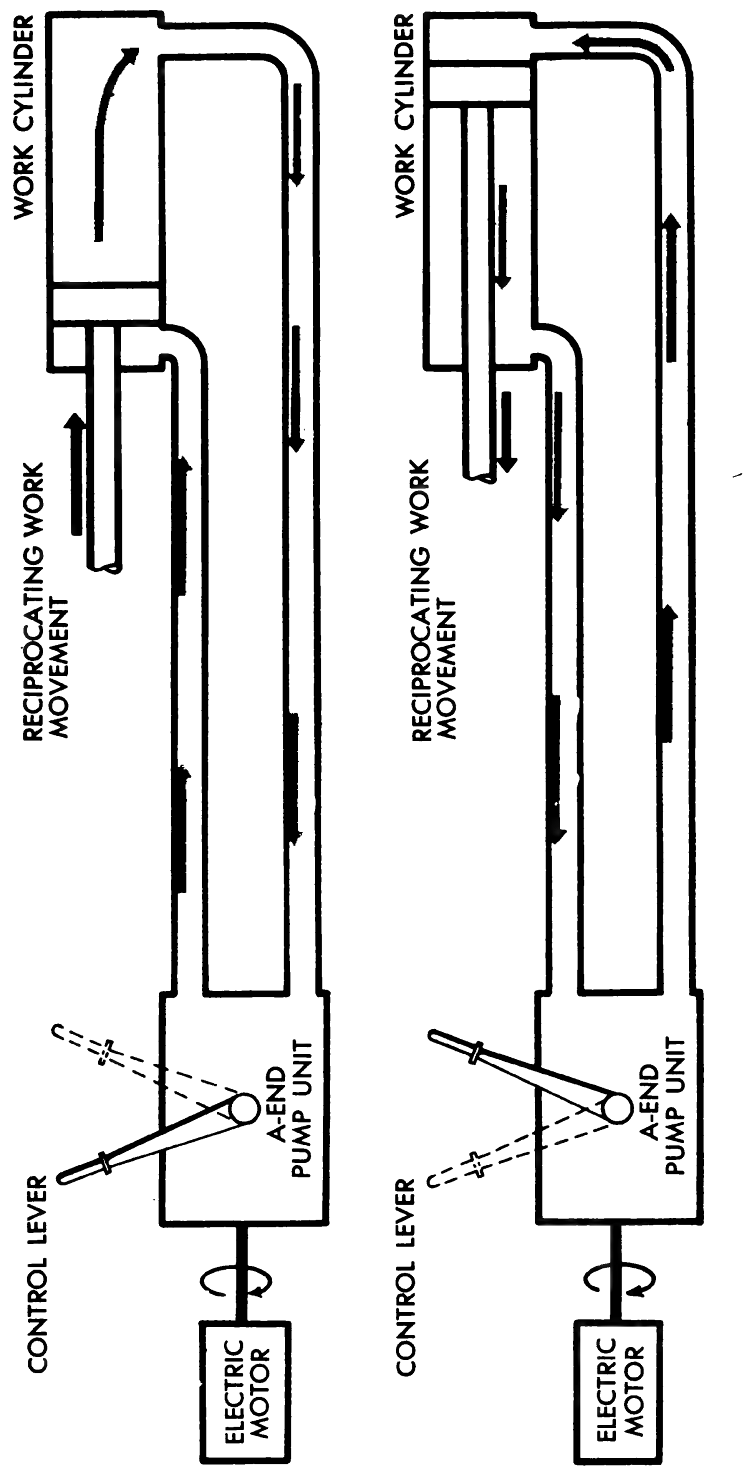
WORK CYLINDER
RECIPROCATING WORK MOVEMENT
CONTROL LEVER
A-END PUMP UNIT
ELECTRIC MOTOR
WORK CYLINDER
RECIPROCATING WORK MOVEMENT
CONTROL LEVER
A-END PUMP UNIT
ELECTRIC MOTOR

Figure 245

work area until sufficient pressure is built up there to open an unloading valve which diverts the liquid from the large pump to the reservoir. High-pressure delivery goes on continuously, but is diluted by low-pressure delivery when pressure in the system drops below the pressure setting of the unloading valve.

(a) *Variable delivery systems with piston and cylinder.* Figure 245 is a simplified block diagram showing the main elements of this kind of system in advance and retract positions.

An electric motor drives a pump which has its tilting plate set by a control mechanism at such an angle that fluid is delivered to one side of a work piston while fluid is being returned from the other side. This takes place because the delivered fluid is under high pressure, while the return fluid is under low pressure, so that the work piston is driven towards the low-pressure end of the work cylinder. In this manner a work operation is performed through mechanical linkages at the outside end of the piston rod.

A reversed positioning of the tilt plate past neutral will reverse the flow of fluid to and from the work cylinder, so that the work piston will be moved in the opposite direction. In this manner a reversed work operation can be performed.

In order to perform the work operation at a slower speed, the control mechanism is set to place the tilting plate more nearly vertical. The tilting plate is set perpendicular to the drive shaft of the pump when no motion at the point of work is desired.

Variable delivery systems with piston and cylinder B-ends are used with powder and projectile hoists and rammers for guns of larger caliber.

(b) *Variable delivery systems with hydraulic motor.* Systems of this kind fall into three sub-groups:

(i) Those under manual control; (ii) Those under automatic control accomplished hydraulically, as in the case of the Ford power drive and the General Electric power drive used for the training and elevation of guns; (iii) Those under automatic control accom-

plished electrically, as in the case of the York and Arma power drives used for the training and elevation of guns.

(i) *Manual control.* Figure 246 is a simplified block diagram showing the main elements of a variable delivery system under manual control for clockwise and for counterclockwise rotation.

An electric motor drives a pump which has its tilting box set by a control mechanism at such an angle that fluid is delivered to one port of a hydraulic motor. At the same time the pump is able to receive fluid from the other port of the motor. The fluid delivered to the hydraulic motor is under high pressure, while the other fluid is under low pressure. The difference in pressure in the two areas of the hydraulic motor forces its pistons to reciprocate. This establishes the one port of the motor as the high-pressure port, and the other as the low-pressure port. It also causes rotation of the cylinder block of the motor, and therefore of the drive shaft to which the cylinder block is attached. In this manner a certain work effect is produced through mechanical linkages at the other end of the drive shaft of the motor.

A reversed positioning of the control mechanism will reverse the flow of fluid to and from the hydraulic motor. Fluid will be delivered to what was previously the low-pressure port, and will be received from what was previously the high-pressure port. The direction of rotation of the hydraulic motor will be reversed, and as a result its drive shaft will be rotated in a direction opposite to that in which it previously rotated. In this manner a reversed work effect is produced at the other end of the drive shaft of the motor.

In order to perform the work operation at a slower speed, the control mechanism is set to place the tilting box more nearly vertical. The tilting box is set perpendicular to the drive shaft of the pump when no motion at the point of work is desired.

Variable delivery systems with hydraulic motor B-ends are used on rammers and on projectile and powder hoists for guns of larger calibers, and on power drives for the train and elevation of guns

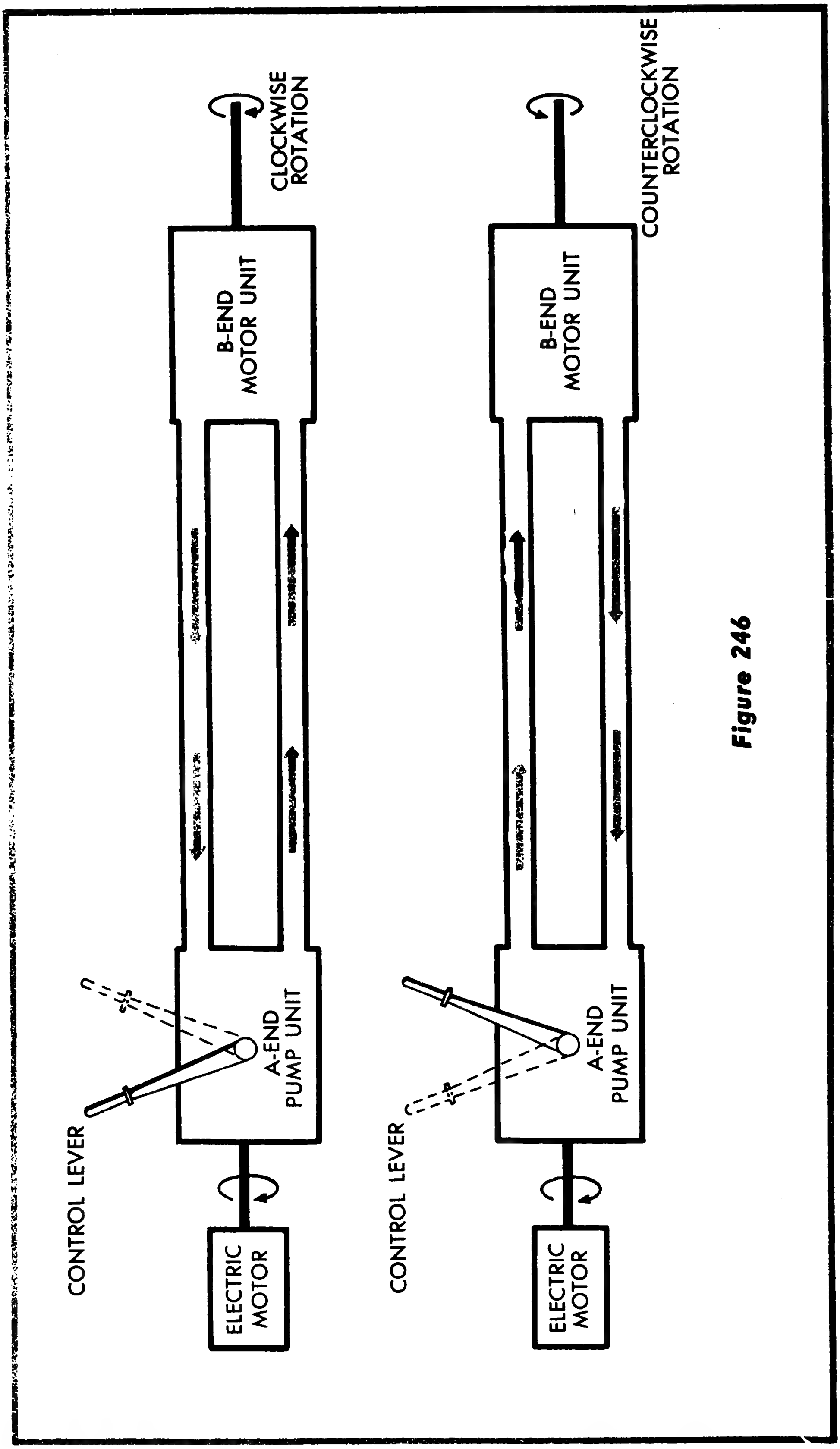
CLOCKWISE ROTATION
B-END MOTOR UNIT
A-END PUMP UNIT
CONTROL LEVER
ELECTRIC MOTOR
COUNTERCLOCKWISE ROTATION
B-END MOTOR UNIT
A-END PUMP UNIT
CONTROL LEVER
ELECTRIC MOTOR

Figure 246

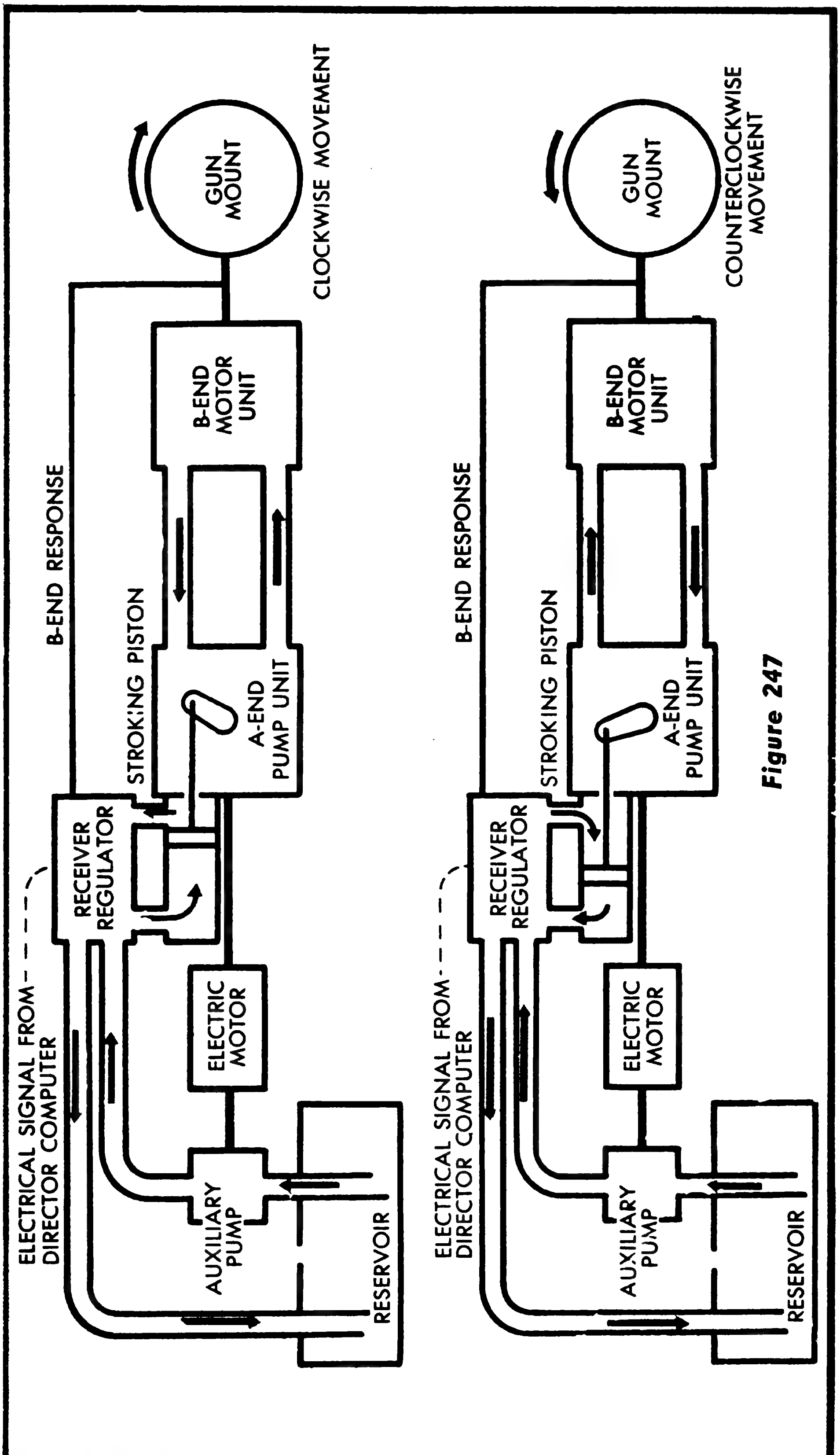
GUN MOUNT
CLOCKWISE MOVEMENT
B-END MOTOR UNIT
B-END RESPONSE
STROKING PISTON
A-END PUMP UNIT
RECEIVER REGULATOR
ELECTRIC MOTOR
ELECTRICAL SIGNAL FROM DIRECTOR COMPUTER
AUXILIARY PUMP
RESERVOIR
GUN MOUNT
COUNTERCLOCKWISE MOVEMENT
B-END MOTOR UNIT
B-END RESPONSE
STROKING PISTON
A-END PUMP UNIT
RECEIVER REGULATOR
ELECTRIC MOTOR
ELECTRICAL SIGNAL FROM DIRECTOR COMPUTER
AUXILIARY PUMP
RESERVOIR

Figure 247

of all calibers. The power drives of many of these guns are also automatically controlled in one or the other of the ways to be described in the two following sections.

(ii) *Automatic control by hydraulic means.* Figure 247 is a simplified block diagram showing the main elements of a variable delivery system under automatic hydraulic control for clockwise and for counterclockwise rotation. This system is used in the Ford and in the General Electric power drives which are employed to position guns in train and in elevation.

There are two principal problems in positioning a gun for fire. One is to get an accurate gun-order signal. This problem is solved by the gun director-computer combination, and does not concern us here. The other problem is to transmit the director signal promptly to the gun, and in such a manner that the position and movements of the gun will be synchronized with signals from the director. This problem is complicated by the fact that movements of the gun tend to fall behind or to overrun director signals, due in part to the lag inherent in transmitting the signals, but mainly to the inertia of the gun. This inertia tends to keep the gun in movement if it is moving, and at rest if it is at rest, whereas the gun-order signal depends primarily on changes in the location of the target. These signals are always changing, not only because of changes in the relative position of the target and the ship, but also because of the roll and pitch to which the ship is subject.

The problem of transforming gun-order signals to mount movement is solved by the power drive of the gun. It consists essentially of a receiver-regulator and a pump-motor combination. The receiver-regulator controls the pump-motor combination, and this in turn controls the movements of the gun.

The receiver-regulator receives an initial electrical gun-order signal from the director-computer, compares it to the existing mount position, and sends an error signal to the hydraulic control mechanism in the regulator. The hydraulic mechanism controls the flow

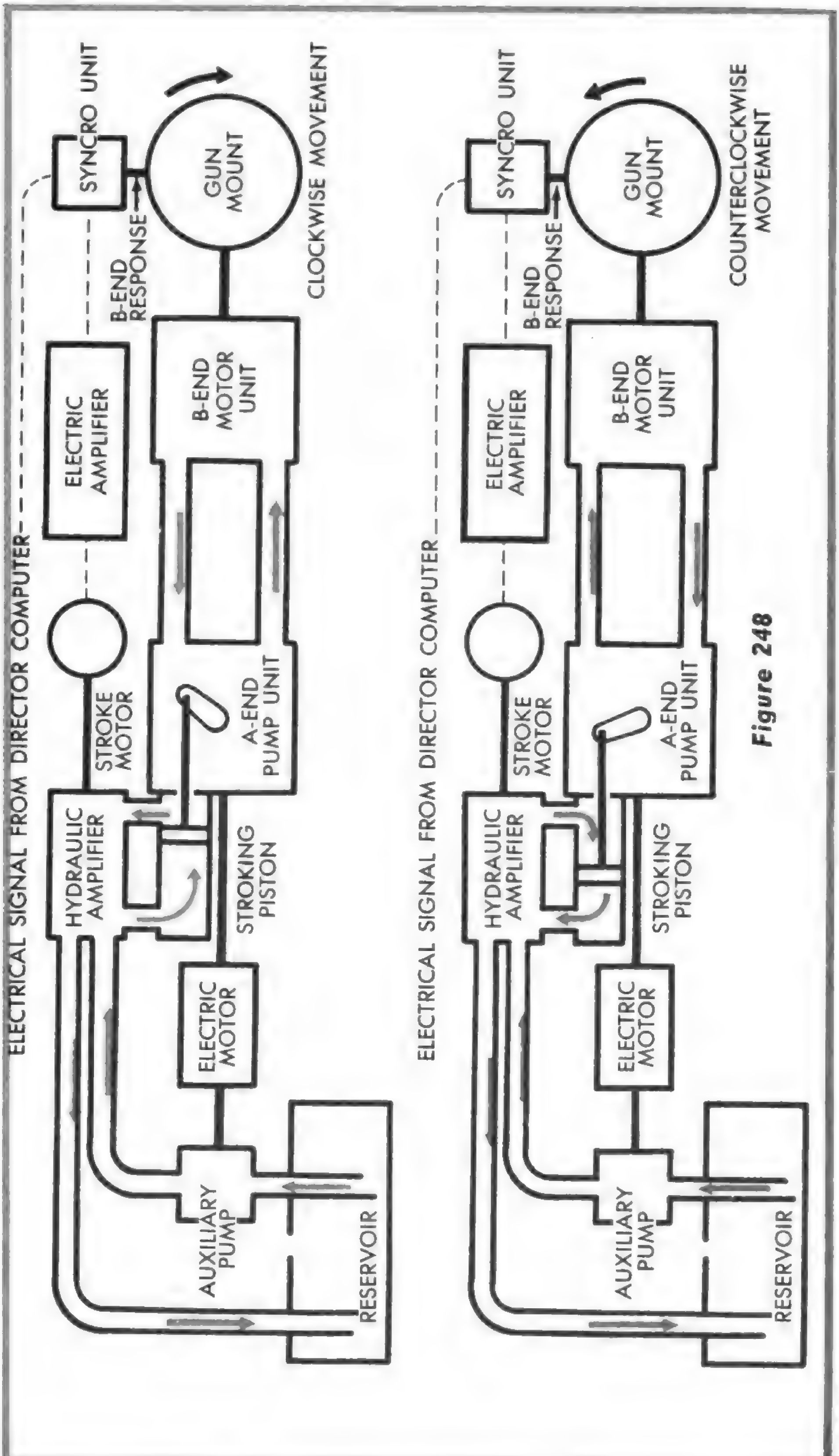
SYNCRO UNIT
GUN MOUNT
CLOCKWISE MOVEMENT
B-END RESPONSE
ELECTRIC AMPLIFIER
B-END MOTOR UNIT
ELECTRICAL SIGNAL FROM DIRECTOR COMPUTER
STROKE MOTOR
A-END PUMP UNIT
HYDRAULIC AMPLIFIER
STROKING PISTON
ELECTRIC MOTOR
AUXILIARY PUMP
RESERVOIR
SYNCRO UNIT
GUN MOUNT
COUNTERCLOCKWISE MOVEMENT
B-END RESPONSE
ELECTRIC AMPLIFIER
B-END MOTOR UNIT
ELECTRICAL SIGNAL FROM DIRECTOR COMPUTER
STROKE MOTOR
A-END PUMP UNIT
HYDRAULIC AMPLIFIER
STROKING PISTON
ELECTRIC MOTOR
AUXILIARY PUMP
RESERVOIR

Figure 248

of fluid which positions the tilting box in the A-end of the pump-motor combination. Its tilt controls the volume and direction of liquid pumped to the B-end, and therefore the speed and direction of rotation of the drive shaft of the B-end. Through mechanical linkages the B-end output shaft moves the gun by an amount and in a direction determined by the signal.

At the same time B-end response is transmitted mechanically to the receiver-regulator, and continuously combined with incoming gun-order signals to give the error between the two. This error is modified hydraulically, according to the rate at which the error is changing, by a system of mechanical linkages and valves in the regulator. When the gun is lagging behind the signal, its movement is accelerated by this means, and when it begins to catch up its movement is slowed down so that it will not overrun excessively.

The hydraulic control mechanism in the regulator positions the tilting box of the A-end exactly after the manner of the constant delivery pump and piston-cylinder system already described in this chapter. The liquid used to move the stroking piston comes from a small auxiliary constant delivery pump driven by the motor which drives the A-end of the main pump-motor combination.

(iii) *Automatic control by electrical means.* Systems of this kind make use of the same general principle of comparing gun position with director-computer gun-order signals to obtain error signals. They also make use of a pump-motor combination controlled by a stroking piston to finally position the gun, in just the same manner as hydraulically controlled systems. In between, however, electrical controls are carried considerably farther than in the system already described. Figure 248 is a simplified block diagram showing the main elements of a variable delivery system under automatic electrical control for clockwise and for counterclockwise rotation. This is the general principle of the systems which are used in the Arma and in the York power drives.

In these systems an electrical gun-order signal is sent by the

director-computer to a synchro unit for comparison with mount position as given by B-end response. The resultant electrical error signal is combined with certain other correction voltages that depend on the amount of error and the rate of change of error, etc. This final error voltage is then sent to an electrical amplifier that amplifies it many times to a value great enough to drive a small electric motor, called the stroke motor. The stroke motor mechanically controls the pilot valve of a hydraulic amplifier which positions the stroking piston of the A-end of the pump-motor combination.

The stroking piston is moved hydraulically by liquid from an auxiliary constant delivery pump in very much the same manner as in the hydraulically controlled system. The modification of the error signal is performed in the hydraulically controlled system by mechanical linkages and valves, however, it is here performed by electrical means. This is the fundamental difference between the two systems.

As already noted, the pump-motor combination moves the mount in a completely familiar manner. The stroking piston determines the tilt on the tilting box; the position of the tilting box determines the volume and direction of flow of fluid to the B-end; this controls the speed and direction of rotation of the B-end drive shaft; this positions the gun; and gun position is mechanically transmitted from the mount to the synchro unit to be electrically combined with gun-order signals from the director-computer to give a continuously changing error signal.

Power drives acting under electrical control as described above are used on 40 mm quad mounts, and on some 6-inch turrets and some modernized 8-inch, 14-inch, and 16-inch turrets.

QUESTIONS

1. How is a hydraulic piston and cylinder used to dissipate the energy of recoil set up when a gun is fired?

2. How are the recoil and counterrecoil systems of a gun related?

3. How is loss of air from the counterrecoil system prevented in some mounts?

4. How is the flow of fluid controlled in constant delivery systems?

5. Name some ordnance uses of constant delivery systems.

6. How is the flow of fluid controlled in variable delivery systems?

7. Upon what does the pressure developed in a variable delivery system depend? What is its relation to the speed at which the pump is rotating?

8. What is the basic difference between variable delivery systems which are controlled hydraulically and systems which are controlled electrically?

9. Where are systems of these two kinds used in naval ordnance?

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(Where there are a number of references under a single heading, the most important are usually italicized.)

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